NASA-CR-194187

p 144

CONTRACT NO. NAS9-17610

THERMACORE, INC.

LANCASTER, PENNSYLVANIA

rel date 3/25/90

FINAL REPORT

PHASE II

SBIR.09.16-1227

MODULAR COLD PLATES FOR HIGH HEAT FLUXES

AUGUST, 1988

PREPARED FOR:

NASA LYNDON B. JOHNSON SPACE CENTER
HOUSTON, TEXAS

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PROJECT SUMMARY

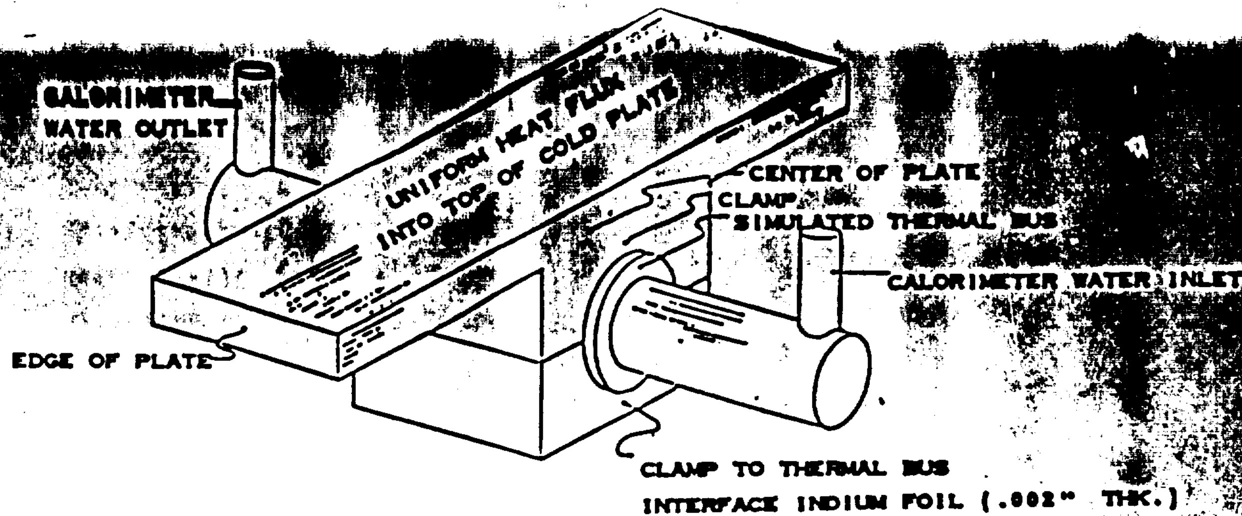
This report describes the work done by Thermacore, Inc., Lancaster, Pennsylvania, for the Phase II, 1986 SBIR National Aeronautics and Space Administration Contract No. NAS9-17610, "Modular Cold Plates for High Heat Fluxes." The work was performed between April 1986 and July 1988.

The principle objective of the Phase II work was to develop a series of modular, high-capacity thermal/mechanical mounting positions (cold plates) and the supporting hardware necessary for use on the Space Station. These cold plates and supporting hardware are intended to provide mechanical support for Space Station experiments/equipment and to transfer waste heat from the experiments/equipment to the thermal bus. The high thermal performance of these cold plates and the supporting hardware was achieved by using heat pipe technology.

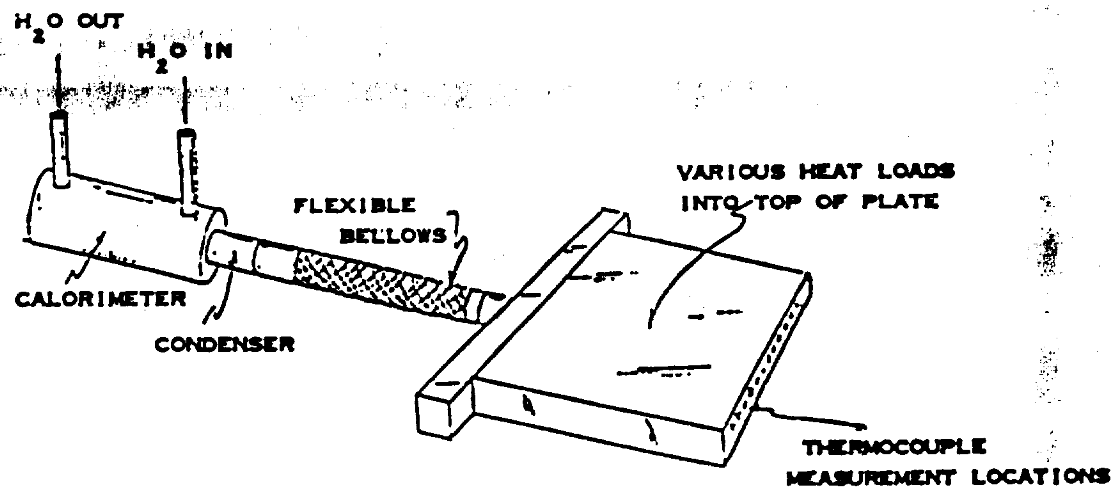
Several pieces of hardware were fabricated and tested: three clamp-on cold plates, two flexible heat pipe cold plates, and two thermal bus receptacles. Figure 1 is a sketch of the hardware and Figure 2 illustrates how the hardware interfaces with the thermal bus. Currently, specific thermal design requirements have not been defined for future Space Station experiments using cold plates. Therefore, with NASA Johnson's concurrence, it was decided that design goals should be established which represent what is believed to be the state-of-the art for cold plate performance. The goals established for each of the fabricated items are listed in Table 1. These design goals were established in Task 1 of the Phase II study.

Transferring heat by conduction from one solid body mechanically clamped to another solid body is hindered by the thermal resistance of the clamped joint. Machining tolerances, differential thermal expansion, and mechanical stress must all be overcome if a successful thermal joint is to be made. This is increasingly difficult with increasing joint size.

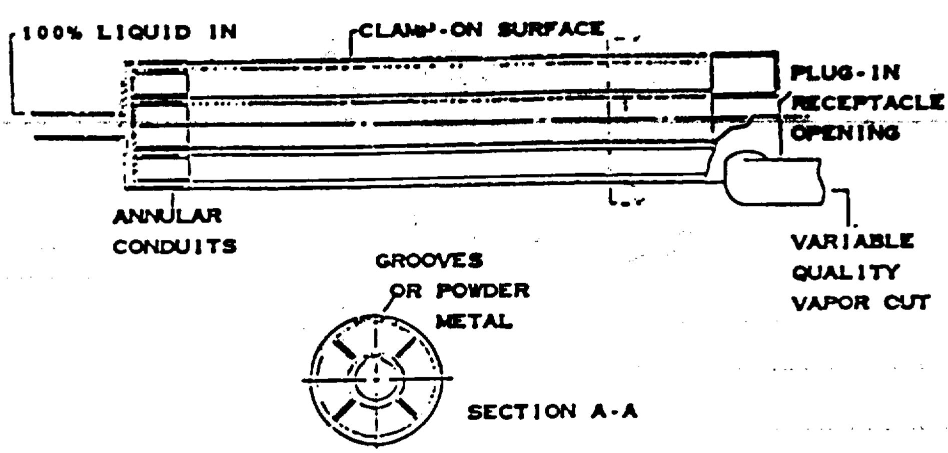
During Task 2, several interface materials were evaluated to determine which materials could reduce the thermal resistance between the mechanically clamped bodies. It was also desirable to be able to disassemble the bodies easily.



Clamp-on cold plate



Flexible heat pipe cold plate



Thermal bus receptacle

Figure 1. Fabricated hardware sketches

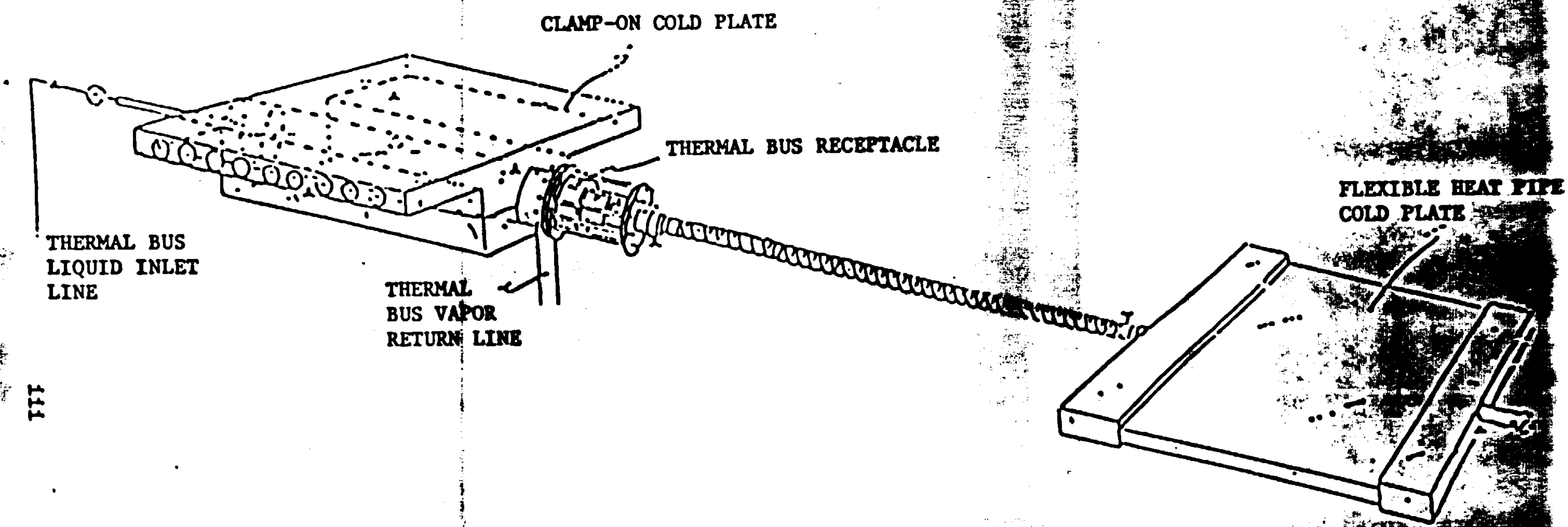


Figure 2. Hardware interfaces with thermal bus

TABLE 1. Preliminary Parameters/Goals

	Clamp-On Cold Plate	Flexible Heat Pipe Cold Plate	Thermal Bus Receptacle
Material:	Aluminum 1100	1100 Aluminum	Aluminum
Working Fluid:	Acetone/Ammonia	Acetone/Ammonia	Ammonia
Mass:	Minimize	Minimize	Minimize
Operating Temperature:	0 - 40°C	0 - 40°C	0 - 40°C
Dimensions:	30 x 30 cm 30 x 60 cm 30 x 90 cm	30 x 30 cm evaporator 45 cm bellows section 30 cm condenser	30 - 50 cm long 5 - 7.5 cm dia.
Temperature Drop:	8 - 10°C	8 - 10°C	---
Total Power Capacity:	2000 - 3000 W	1000 W	3000-4000 W

Indium foil placed between the two bodies prior to clamping proved to be an excellent thermal interface material for large area joints. The low tensile strength of the material allows the metal to flow, when clamped, and fill any imperfections of the clamping surfaces. For clamped interfaces, as shown in Figure 2, a 0.0019 inch thick Indium foil was demonstrated to have less than 1°C temperature drop at 7 W/cm^2 over a 239 cm^2 cylindrical joint clamped at 350 psi.

During the third task of the program, the clamp-on cold plates, designed during the Phase I effort, were fabricated and tested. A clamp-on cold plate is a device which can be remotely outfitted with an experiment or piece of equipment and then physically clamped onto the thermal bus. Figure 1 shows a schematic of the clamp-on cold plate. The waste heat generated by the experiment/equipment is transferred to the thermal bus working fluid. The advantages of this type of cold plate are remote outfitting of experiment/equipment, modularity, "off the shelf" capabilities, flexibility, and contingency. Three cold plates were successfully fabricated, tested, and delivered to NASA JSC.

Task 4 extended the clamp-on cold plate concept to add another degree of freedom to the application of the cold plate technology. A flexible heat pipe cold plate is a device which can be remotely outfitted with an experiment or piece of equipment and then physically "plugged in" to a thermal bus receptacle. Figure 1 shows a schematic of the flexible heat pipe cold plate. The waste heat generated by the experiment/equipment is transferred to the condenser section of the heat pipe that is inserted into the thermal bus receptacle (see Figure 2). The heat is then conducted across a tapered male-female joint and transferred to the thermal bus working fluid. The advantages of this type of cold plate are remote outfitting of experiment/equipment, modularity, "off the shelf" capabilities, flexibility, and contingency. Two flexible heat pipe cold plates were fabricated, tested, and one was delivered to NASA JSC.

The fifth task of the program included the design, fabrication, and test of thermal bus receptacles. A thermal bus receptacle is a device that is integral to the thermal bus and is a location of enhanced heat transfer. The receptacle is a device that a clamp-on cold plate can be attached to or a flexible heat pipe cold plate can be "plugged in."

Figure 2 is a schematic of a thermal bus receptacle showing the versatility of the receptacle, clamp-on cold plate, flexible heat pipe cold plate system.

The thermal bus receptacle utilizes the thermal bus working fluid and a heat pipe wick structure to enhance the heat transfer between the cold plate and the receptacle. Two thermal bus receptacle concepts were generated, fabricated, tested and delivered to NASA JSC.

The sixth and final task of the program included assembling and testing the hardware as a heat transport package. The design, fabrication, and testing of the heat pipe hardware demonstrated Thermacore's ability to manufacture heat transfer devices that could be used on the Space Station. The heat transfer package (clamp-on cold plate, flexible heat pipe cold plate, thermal bus receptacle) demonstrated its potential as a modular, high capacity thermal/mechanical heat transfer system.

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1.0 INTRODUCTION

The thermal bus is one of the key systems contributing to the potential versatility of the Space Station. In an analogy between thermal and electrical circuits, the Thermal Bus provides a centrally located heat sink to which heat can flow from sources distributed throughout the Space Station. To carry the analogy further, this effort was focused on the development of the wall outlets and line cords that will make it possible to use the Thermal Bus to its fullest extent.

Thermal bus receptacles developed by Thermacore under this Phase II contract are intended to be placed into the existing Space Station thermal bus at predetermined locations. The thermal bus receptacles are the wall outlets. Once in position, the receptacles serve to transfer waste heat from equipment and experiments from the Space Station through the thermal bus receptacles and out to the Space Station radiators where the heat can be radiated to deep space.

Clamp-on cold plates and flexible heat pipe cold plates developed by Thermacore under this Phase II contract will serve as heat transfer supporting hardware. These components are the line cords. Equipment and experiments on the Space Station will be mounted directly to the evaporator of these cold plates. This work can be accomplished inside the Space Station because of the modularity of the heat pipe components. From there, the experiments and equipment mounted onto the heat pipe components can be taken out to the thermal bus and mounted directly onto or into the thermal bus receptacles. This modular type of heat transfer system will reduce the time required and risk associated with working outside of the habitat and work modules.

Thermacore developed heat transfer components to expand the versatility of the thermal bus, these components include, two separate thermal bus receptacles (wall outlets); three clamp-on cold plates to be mounted on the outside of the thermal bus receptacles (line cords); and two different flexible heat pipe cold plates (line cords) to be plugged into the thermal bus receptacles.

Six technical tasks were included as part of this Phase II program.

- Task 1 - Requirements: Establish thermal and mechanical requirements for high performance modular cold plates (Table 1).

- Task 2 - Clamped Joint Interface Delta-T: Demonstrate a mechanically clamped interface joint between the cold plate and the thermal bus that has a low delta-T and is easily removable from the thermal bus. (Section 3.0)

- Task 3 - High Performance Cold Plates: Fabricate and demonstrate the performance versus delta-T for the 30 x 30 cm, 30 x 60 cm and 30 x 90 cm cold plates designed in the Phase I effort. (Section 4.0)

- Task 4 - Flexible Heat Pipe Cold Plate and Thermal Bus Coupler: Design, fabricate and demonstrate a flexible heat pipe cold plate which can be coupled to the thermal bus. (Section 5.0)

- Task 5 - Thermal Bus Alternatives: Modify existing single phase cooling loop design and/or design a two phase heat pipe type thermal bus. (Section 6.0)

- Task 6 - Consolidation of Heat Acquisition - Heat Transfer System: Assemble the hardware, demonstrated individually, and test the entire system. (Section 7.0)

The remainder of this report separately covers each of the above tasks. The discussion will include: requirements for each task, design, fabrication and testing procedures, test results, conclusions and recommendations.

2.0 CONCLUSIONS

The principle objective of this Phase II program was to design, fabricate, and test a series of modular, high capacity thermal/mechanical cold plates and the supporting hardware necessary for use on the Space Station. This objective was successfully met in every aspect. For example, a series of three clamp-on cold plates were fabricated and demonstrated to carry up to 1500 W at 10°C overall delta-T, operating up to 4" adverse tilt (> 6° angle), and greater than 6 W/cm^2 heat input without a sign of dryout. The specific results and conclusions for each of the fabrication and test tasks are shown below.

2.1 CLAMPED JOINT INTERFACE DELTA-T

Indium foil met the thermal performance requirement, demountability requirement, and the off-gassing requirement for a low thermal resistance clamped interface joint. Indium foil exhibited a delta-T of <1°C at 7 W/cm^2.

2.2 CLAMP-ON COLD PLATES

The 30 x 30 cm and 30 x 60 cm plates performed as calculated with acetone and ammonia. The 30 x 90 cm plate performed as expected with acetone, however, it appears that it did not maintain primed arteries with ammonia.

A uniformly distributed heat load is the best loading for the present design. However, strip loads at heat flux levels up to 6 W/cm^2 are also easily accommodated provided the loads are perpendicular to the heat pipes.

Each cold plate design is required to transport power with a temperature drop of 10°C or less. Tests show that temperature drop is relatively independent of cold plate size and is more a function of the clamp design. Most of the temperature drop occurs because of the heat pipe delta-T of condensation and the delta-T of conduction through the clamping mechanism and interface gap.

Higher power carrying capabilities are easily accommodated by the present heat pipe design, however, a larger condenser area and a lower thermal resistance clamp and interface are required to meet the 10°C overall delta-T limit.

2.3 FLEXIBLE HEAT PIPE COLD PLATES

The FHPCP with 2.54 cm diameter vapor space performed as expected with acetone, however, it appears that it did not maintain primed arteries with ammonia. The depriming of the arteries is possibly due to flux material introduced during the sintering process. This could affect the wetting angle and lead to poor capillary performance.

The FHPCP with a 2.54 cm diameter vapor space, using acetone as the working fluid, performed best with the heat loading uniform over the evaporator area. Strip loads normal to the heat pipes were also successfully demonstrated.

The FHPCP with a 2.54 cm diameter vapor space, using ammonia as the working fluid performed best with all heater geometries except for the heater positioned around the perimeter of the evaporator.

The FHPCP with a 1.59 cm diameter vapor space did not operate as designed. The poor performance was determined to be a capillary limit caused by a large pressure drop in the artery manifold which supplies the 16 evaporator pipes with liquid drawn from the condenser.

2.4 THERMAL BUS RECEPTACLES

The Flow Through (TBR I) and Reservoir (TBR II) Thermal Bus receptacle design are capable of transferring the waste heat from the clamp-on or plug-in cold plates to the thermal bus.

Strict control over the ammonia liquid level in the reservoir of TBR II is essential to provide uniform heat rejection.

Closely matched surfaces and surface finishes of 16 rms or better are required between the FHPCP condenser and the thermal bus receptacles in order to minimize the interface resistance and delta-T.

2.5 CONSOLIDATION OF HEAT ACQUISITION HEAT TRANSFER SYSTEM

The consolidation of the heat transfer components as a unit heat transfer system demonstrated the concept in principal; however, the performance was less than the design goals. Because the components were fabricated, tested, and modified as separate components, the interface joints were not as good as they originally were intended to be. The result was a lower than calculated heat transfer capability which further reinforces the previous conclusions that good interface joints are critical to good performance in this type of a modular system.

A Flow Through Cold Plate capable of carrying greater than 3000 watts with a less than 1°C delta-T at 3000 W, was also fabricated and tested. The flow through cold plate was an additional task beyond the original scope of work. This flow through cold plate used the technology developments from the clamp-on cold plate, flexible heat pipe cold plate and thermal bus receptacles. The end result is a low mass heat pipe cold plate with a high heat transfer capacity at a corresponding low delta-T.

3.0 CLAMPED JOINT INTERFACE DELTA-T

In order for the cold plate hardware developed under this program to be successfully clamped to a thermal bus loop, an interface material with good thermal conductivity was necessary to enhance the heat transfer performance across the gap of the a mechanically clamped joint.

This work began in February, 1985, under a Phase I SBIR NASA JSC Contract No. NAS9-17280, "Modular Cold Plates for High Heat Fluxes," and was completed under this Phase II Program.

The principle objective of the interface material task was to test a variety of materials in an effort to find a low delta-T, easily demountable, and low outgassing clamped joint interface material. Several types of materials were tested, including: thermally conductive epoxies, thermally conductive rubber pads, metal foils, and various other low vapor pressure materials.

The materials are to be used between the thermal bus and the clamp-on cold plates (Section 4.0) or the thermal bus receptacle (Section 6.0) and the flexible heat pipe cold plates (Section 5.0). Sections 3.1 - 3.4 describe the interface requirements, the test matrix, the test procedure, and the results and conclusions.

3.1 REQUIREMENTS

The first task of the mechanically clamped interface task was to determine the minimum requirements for the clamped joint. Based on the dimensions and requirements for the clamp-on cold plates, flexible heat pipe cold plates, and thermal bus receptacles, the following requirements were selected:

Interface Delta-T	1-3°C at 7 W/cm^2
Demountability	Manually demountable
Material Outgassing	Low outgassing (low vapor pressure)

3.2 TEST MATRIX

The second task of the interface materials investigation was to design and fabricate a test vessel that was representative of an actual clamped joint. For this program, the clamped interface was a cylindrical geometry approximately 7.62 cm in diameter. A photograph of test hardware is shown in Figure 3.

Figure 4 is a cross-sectional assembly drawing of the test vessel. The materials tested were applied at the thermal interface between the simulated thermal bus and the simulated cold plate. The simulated cold plate was clamped onto the simulated thermal bus with bolts. Heat was applied to the outer diameter of the simulated cold plate through electrical resistance heaters. The heat was transferred radially inward, where it was removed by the cooling water. The flow chamber insert was used to increase the heat transfer coefficient at the simulated thermal bus/cooling water interface. Thermocouples located in the test fixture and in the cooling lines were used to determine the delta-T across the thermal interface and to calculate the total power transferred.

Four groups of materials were tested: metal foils, epoxies, thermal pads, and miscellaneous materials. The specific materials are listed in Table 2.

3.3 TEST PROCEDURE

The following test procedure was used to collect the thermal resistance data for a clamped joint.

A) Apply the thermal resistance material to the inner diameter of the simulated thermal bus.

B) Place the two halves of the simulated cold plate onto the simulated thermal bus and bolt the two halves together.

C) Torque the bolts to approximately 14 ft-lbs. For this particular test, a torque of 14 ft-lbs results in a clamping pressure of approximately 350 psi.

D) Attach an electrical resistance heater to the outer diameter of the simulated cold plate followed by two inches of Kaowool insulation blanket.

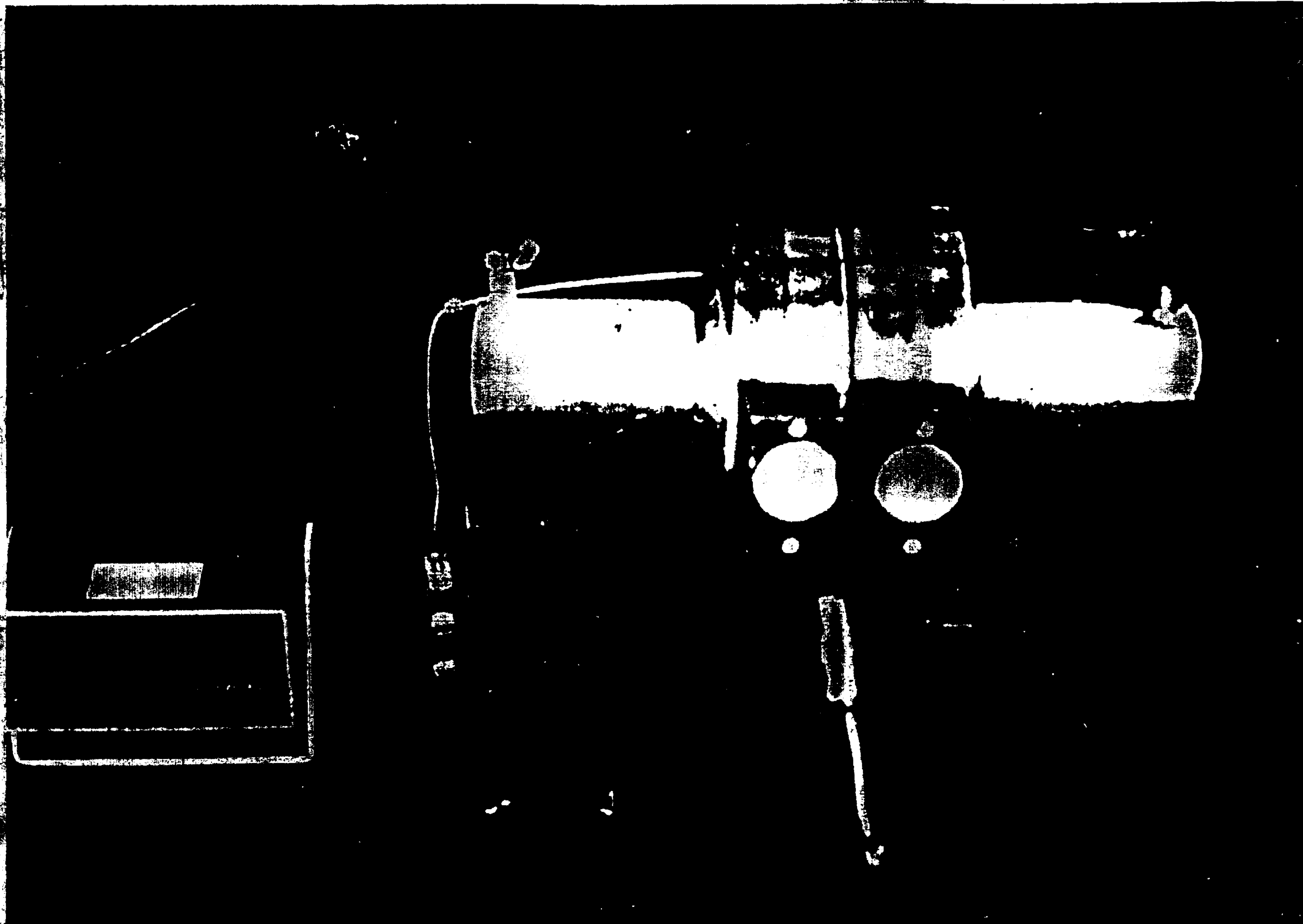

Figure 3.—Thermal resistance test (Wet)

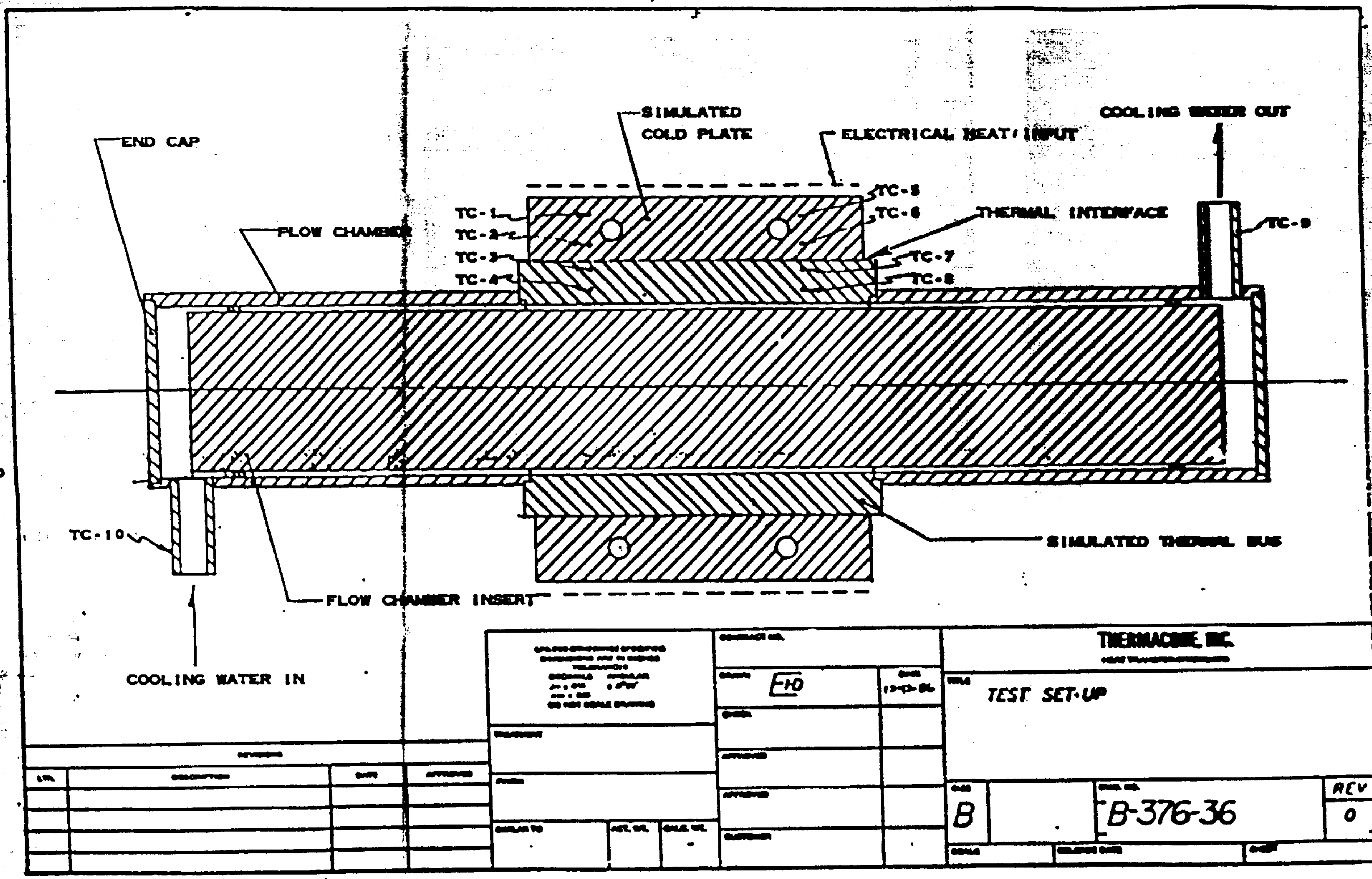

Figure 4. Cross-sectional assembly drawing of thermal resistance test vessel

TABLE 2. Interface Materials Test Matrix

Foils	Miscellaneous
Aluminum Foil (.001")	Molybdenum Disulfide
Indium Foil (.0019")	Silicon Vacuum Grease
	Thermatrace NH
Epoxies	Watlube
Tra-Con Tra-Duct 2902	Silicon Oil
Emerson & Cuming Stycast 2850 Ft	Emerson & Cuming TC4 Thermal Grease
Magnolia Compound 81-090	Grapho 507 Graphite
Thermoset 340 Resin	Air Between Polished Surfaces (8 RMS)
	Air Between Rough Surfaces (64 RMS)
Thermal Pads	
Coolsil Pad 1867 & Spray Adhesive	
Coolsil Pad HS-007-AC	
Coolsil Pad 1867	
Q-Pad	

E) Turn the cooling water on and turn on the electrical heater. For this test vessel, the power in was initially set at approximately 250 watts.

F) Allow the test vessel temperatures to reach steady state and record the thermocouple readings into the laboratory notebook.

G) Increase the electrical power 250 watts and repeat F-G up to approximately 2000 watts.

3.4 RESULTS AND CONCLUSIONS

Figures 5-8 are the results of the test matrix. The figures are plots of heat flux (W/cm^2) versus delta-T (°C). Several materials can meet or exceed the heat flux/delta-T requirement of 1-3°C at 7 W/cm^2. These materials include: Indium foil (.0019"), Tra-Con Tra-Duct 2902 Epoxy, and Emerson & Cuming TC4 Thermal Grease.

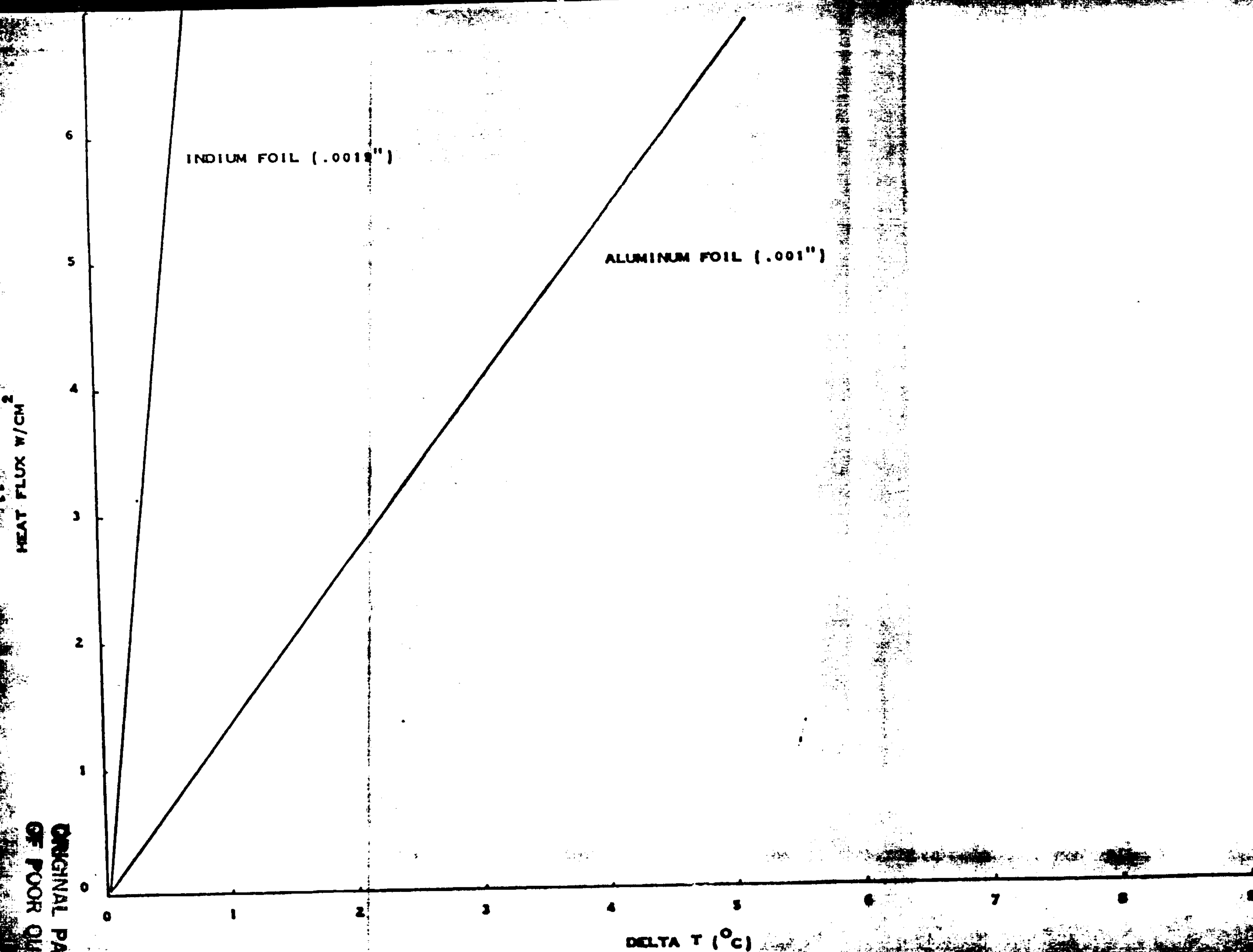

Figure 5. Foils

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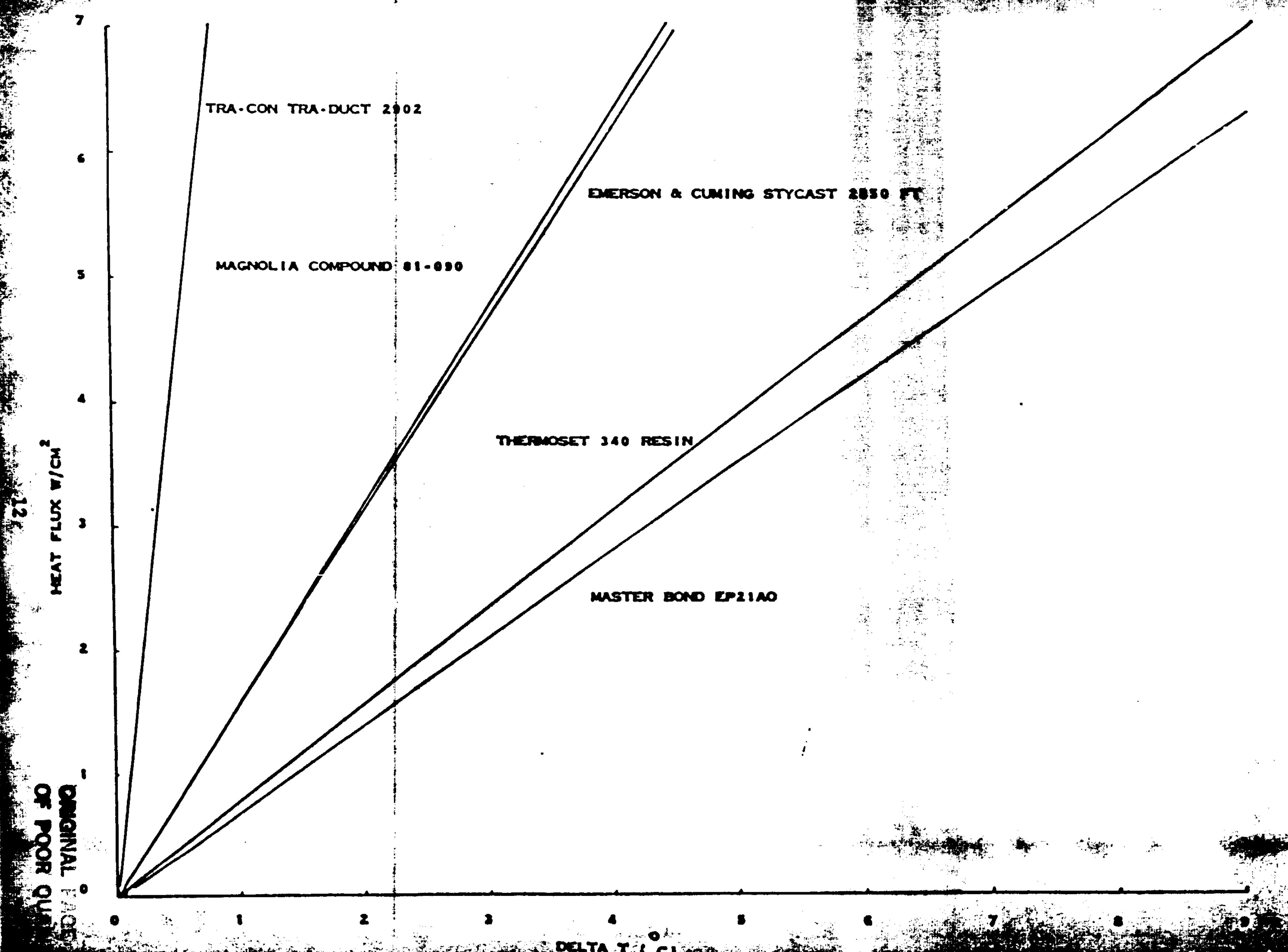

Figure 6. Epox

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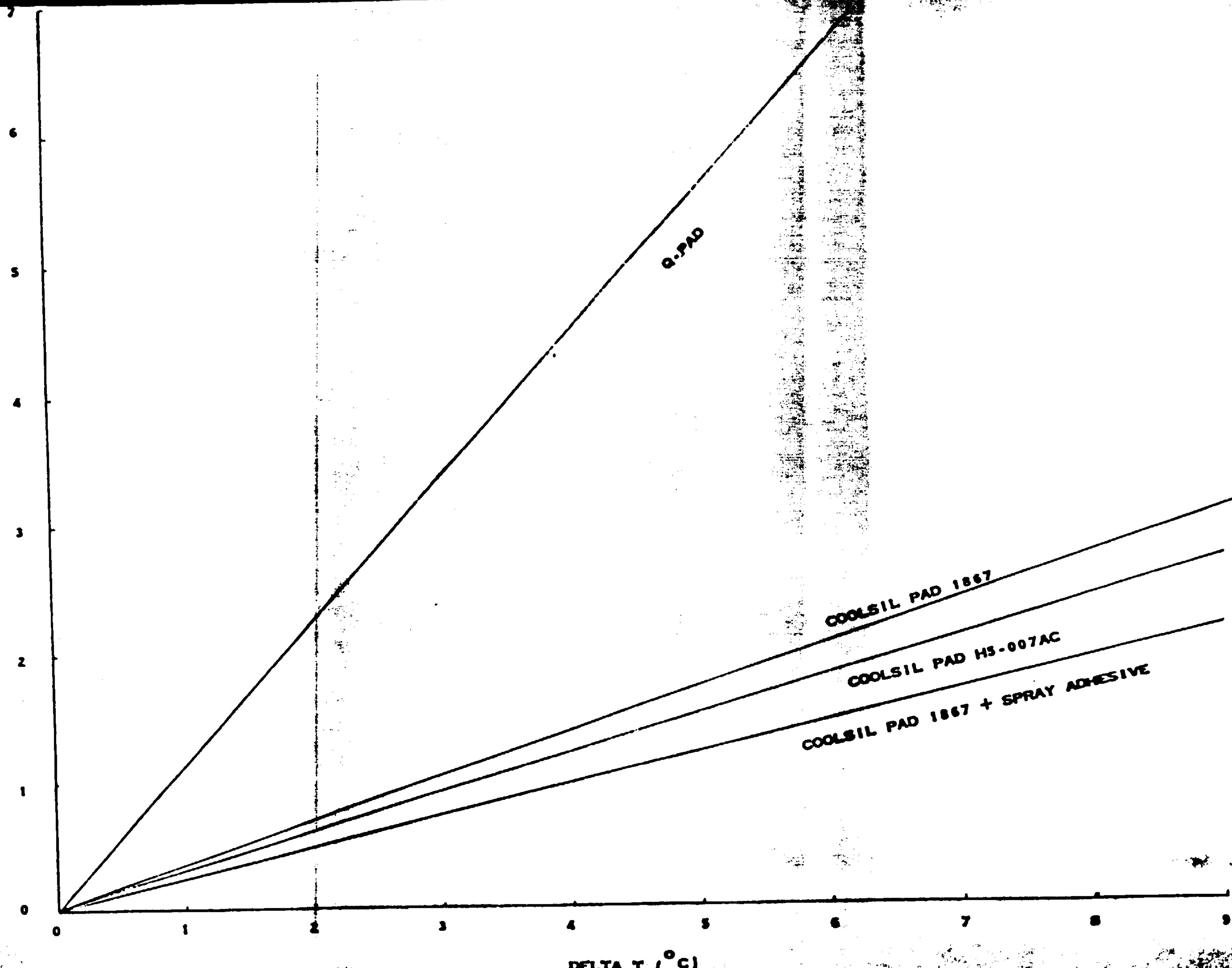


Figure 7. Thermal pads

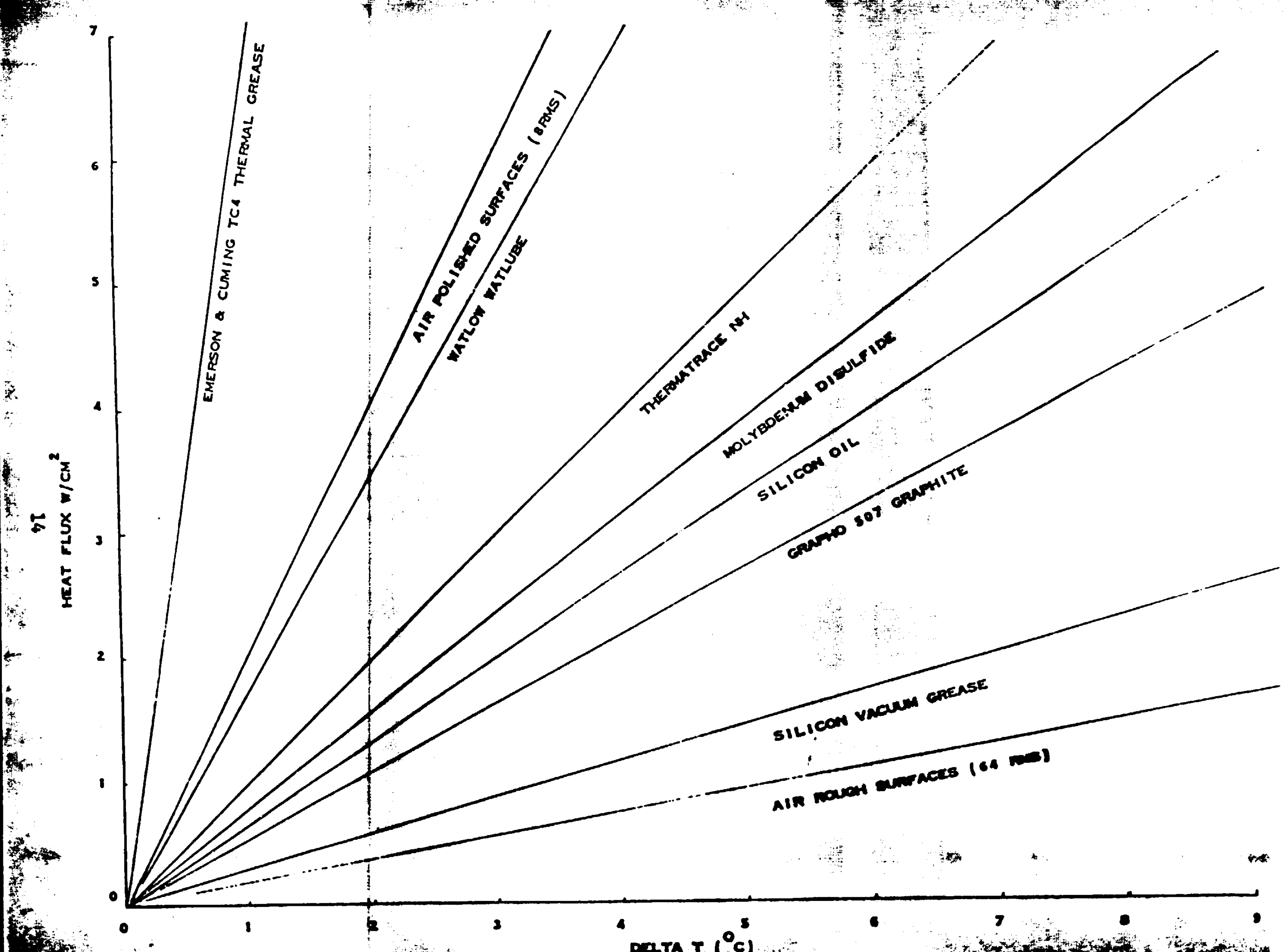


Figure 8 Miscell

Of these materials, only Indium foil can meet the thermal performance requirement, the demountability requirement, and the offgassing requirement. The epoxy is not easily demountable. The epoxy must be physically broken (with excessive force) or thermally broken down at a temperature which causes excessive ammonia vapor pressure. The thermal grease offgasses rapidly in a high vacuum environment and "dries out" causing a decrease in thermal performance with time.

The thermally conductive rubber pads were the most convenient material to work with but are not manufactured thin enough to get the required performance. Q-pad is 0.006" ± 0.001" thick and has a delta-T of 6.5°C at 7 W/cm². Pads with thicknesses less than 0.003" would lower this delta-T to below 3°C at 7 W/cm². Therefore, until these pads are available, Indium foil is the only material tested that meets the requirements.

Indium foil has a melting point of 156°C, a thermal conductivity of 85.4 W/m°c, and a tensile strength of 380 psi. The melting point indicates that the foil will be a solid in the operating temperature range of 0 to 40°C. Solid metals generally have low vapor pressure which implies low outgassing. The thermal conductivity is 20 to 50 times higher than epoxies and thermal greases. And, the low tensile strength allows the metal to creep and fill any imperfection on the clamped surfaces. For these reasons, Indium foil was selected as the interface material for the clamp on cold plates.

4.0 HIGH PERFORMANCE COLD PLATES

The three clamp-on, modular cold plates, designed and performance predicted during the Phase I effort, were fabricated and tested. The fabrication, processing, and performance testing information gained was intended to verify the design principles used and to provide a data base of performance as a function of heat load placement. Sections 4.1 through 4.6 detail the work effort completed in Phase II.

4.1 REQUIREMENTS

During the Phase II effort, no specific thermal design requirements for clamp-on cold plates were defined by NASA for future Space Station experiments. Accordingly, Phase B Space Station criteria were used to set the applicable operating temperature range and allowable delta-T's. Based on the operating temperature and allowable delta-T, preliminary design calculations were performed using state-of-the-art values for the design parameters. The result of these calculations was to set the power transfer capability requirement. Material selection and physical dimensions were a direct result of the Phase I design effort. Table 3 is a listing of the clamp-on cold plate design parameters/requirements.

TABLE 3. Clamp-On Cold Plate Parameter/Requirements

PARAMETER	MAGNITUDE
Structural Material	1100 aluminum
Working Fluid	Ammonia/Acetone
Mass	Minimize
Operating Temperature	0 - 40°C
Dimensions of Plate	30 x 30 cm
	30 x 60 cm
	30 x 90 cm
Temperature Drop	8 - 10°C
Power Transfer Capability	2000 - 3000 W

4.2 DESIGN

The design effort for the three clamp-on cold plates was performed during the Phase I effort. The sizing calculations, optimization schemes, and performance predictions are all described in detail in the Final Report for NASA Contract Number NAS9-17280 "Modular Cold Plates for High Heat Fluxes." Figure 9 is a sketch of the selected design. However, a change was made to the proposed wick structure. The proposed, two artery "D" shaped wick structure was changed to a single, pedestal artery.

The modification was made after several single heat pipe tests showed that vapor generated at the wall/wick interface was penetrating the arteries and preventing the return of liquid to the evaporator. For these cold plates to operate as designed, the wick structure selected must be all of the following: high performance (sintered powder metal, arterial), prime easily (arteries), and resist vapor penetration caused by the boiling at the wall/wick interface (arteries). Sintered powder metal pedestal arteries are inherently high performance, and have been demonstrated to prime themselves. The only remaining problem was to prevent vapor penetration into the arteries.

For all practical purposes, boiling is an unavoidable phenomenon in ambient temperature heat pipes. The boiling itself does not greatly affect the performance of the wick, but rather the vapor generated penetrates the arteries and causes them to deprime, which in turn, causes the heat pipe to dryout.

Tests run under the US Air Force Contract #F33615-84-C-3415, "High Performance Aluminum Heat Pipe Development," indicated that the nucleation sites are located in the first few layers of powder sintered to the walls. Therefore, to keep the vapor generated from penetrating the arteries, a wick structure needs to be designed with the pressure drop to the vapor space less than the pressure drop to the artery. This type of wick structure would force the vapor generated to circumvent the artery.

Figure 10 is a concept drawing of two wick structures which were developed to verify the above hypothesis and to demonstrate on the bench a high performance, easily primed, boiling resistant artery heat pipe.

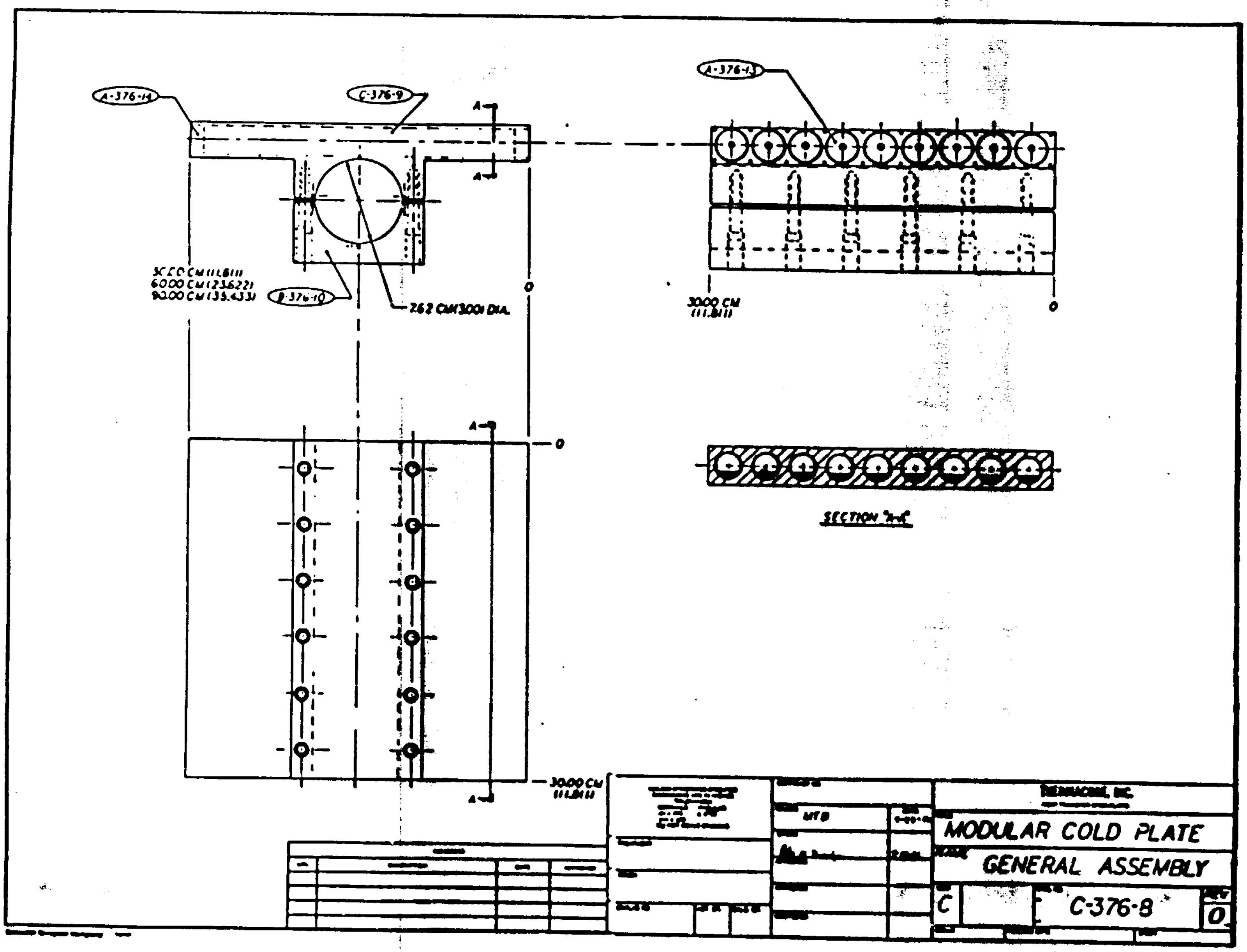
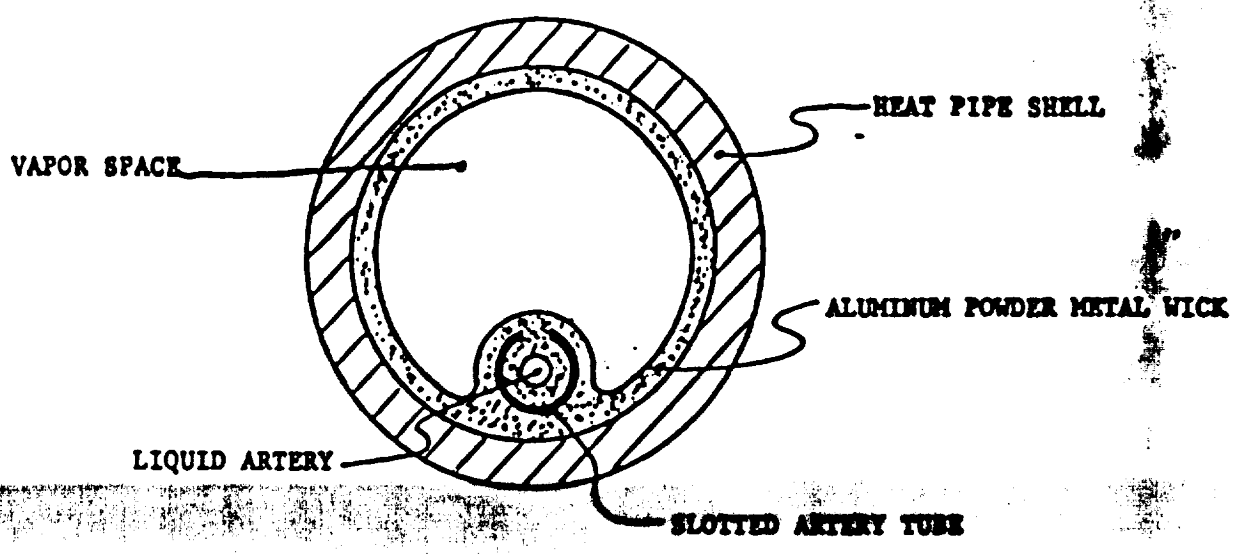


Figure 9. Modular cold plate



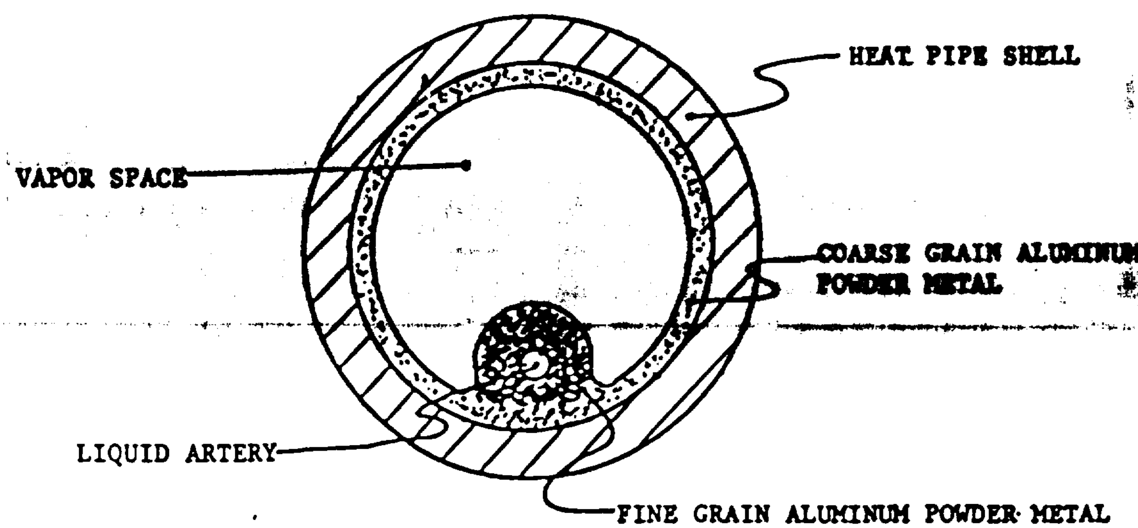


Figure 10. Boiling resistant artery designs

A three foot long heat pipe with a combination of grain sizes (course circumferential wick-fine artery wick) wick structure was fabricated and tested using acetone as the working fluid. The heat pipe bench test set up is shown in Figure 11. The heat pipe was oriented with the evaporator above the condenser (against gravity). Tests were run at horizontal, one, two, three, etc. inches against gravity and the power was increased until the evaporator dried out. Figure 12 shows data collected for the combination grain size boiling resistant wick structure heat pipe using acetone as the working fluid. This particular heat pipe performed better than predicted.

Based on the boiling resistant artery effort the wick structure shown in Figure 13 was selected and has the following features:

- Pedestal Artery Design
- Coarse powder circumferential wick
- Fine powder artery wick
- Slotted aluminum artery liner

The coarse powder circumferential wick has good permeability/pore radius properties which gives the heat pipe high performance. The fine powder artery and the slotted artery liner impose a high pressure drop barrier around the artery to prevent vapor penetration and possible depriming of the artery. The smooth inner wall of the artery tube may also delay the incipience of boiling inside the artery. Also, the condensation delta-T in the plate is proportional to the wick thickness at the condensation site. The pedestal artery design is thinner than the "D" shaped design in the condensation area. And, finally, tests with acetone have shown that the pedestal artery design is capable of priming on earth; and therefore, the "D" shaped wick design is not necessary.

The vapor penetration (boiling) resistant artery concepts developed under this program were patented.

Because of the developing nature of the boiling resistant arteries, the cold plate design was not re-optimized based on the selected wick structures. However, the thermal performance of the as fabricated cold plates was predicted. The calculated performance was based on evaporator, adiabatic, and condenser areas as shown in Figure 14. The evaporator area is considered to be the entire circumference of the heat pipe not above

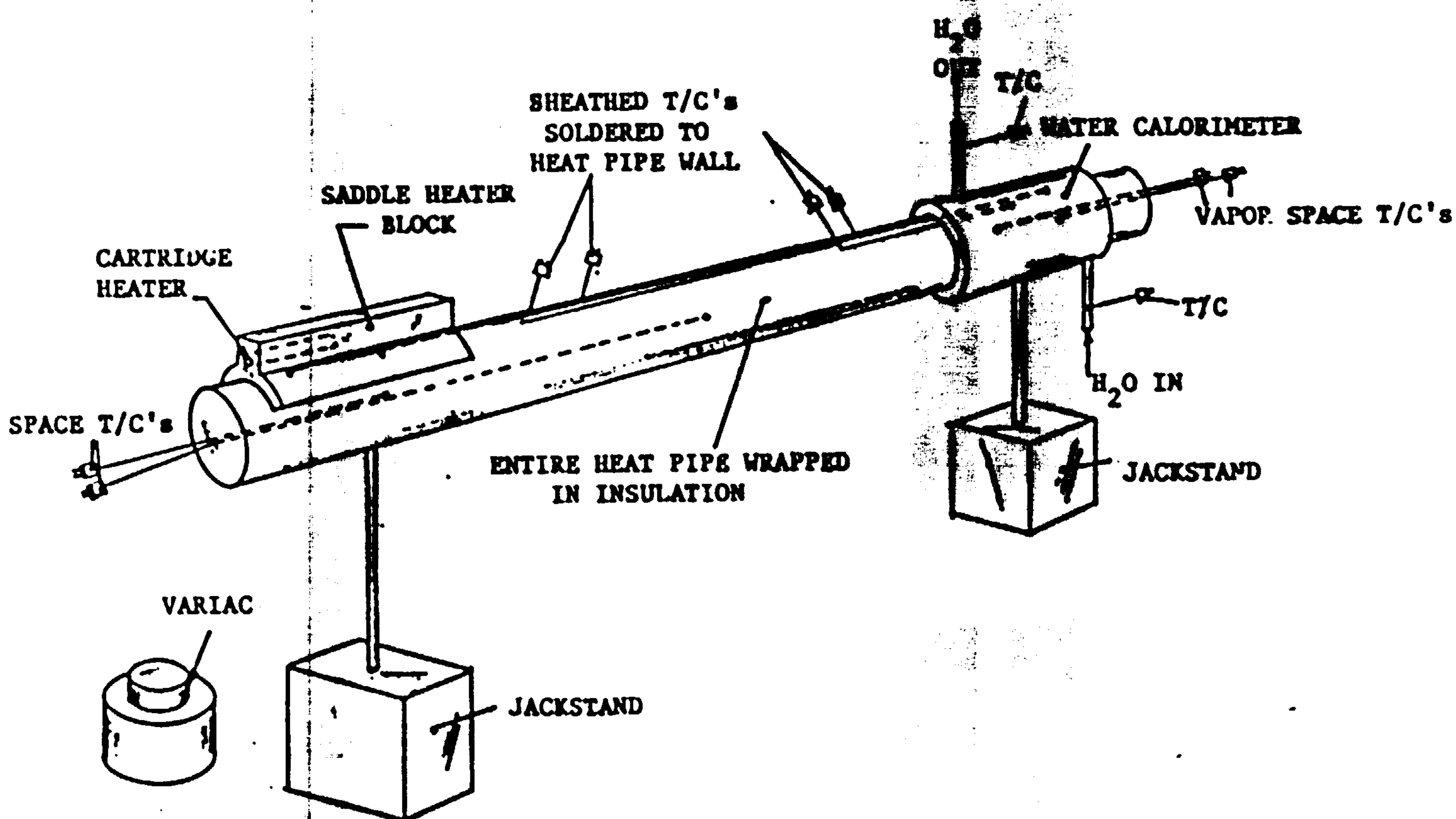


Figure 11. <u>Heat pipe test set-up</u>

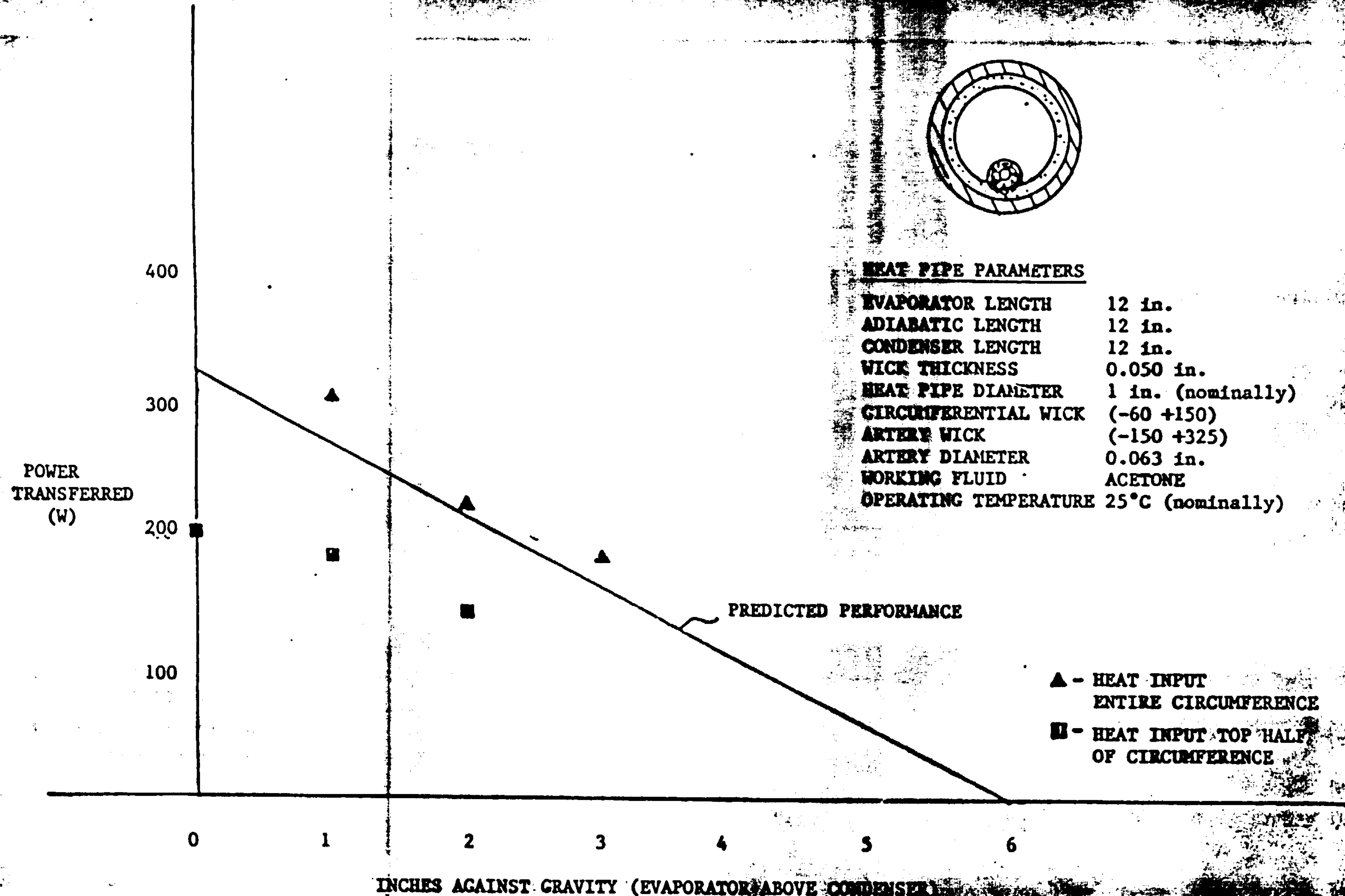


Figure 12. Power vs. inches against gravity for coarse powder circumferential, fine powder artery boiling resistant wick structure

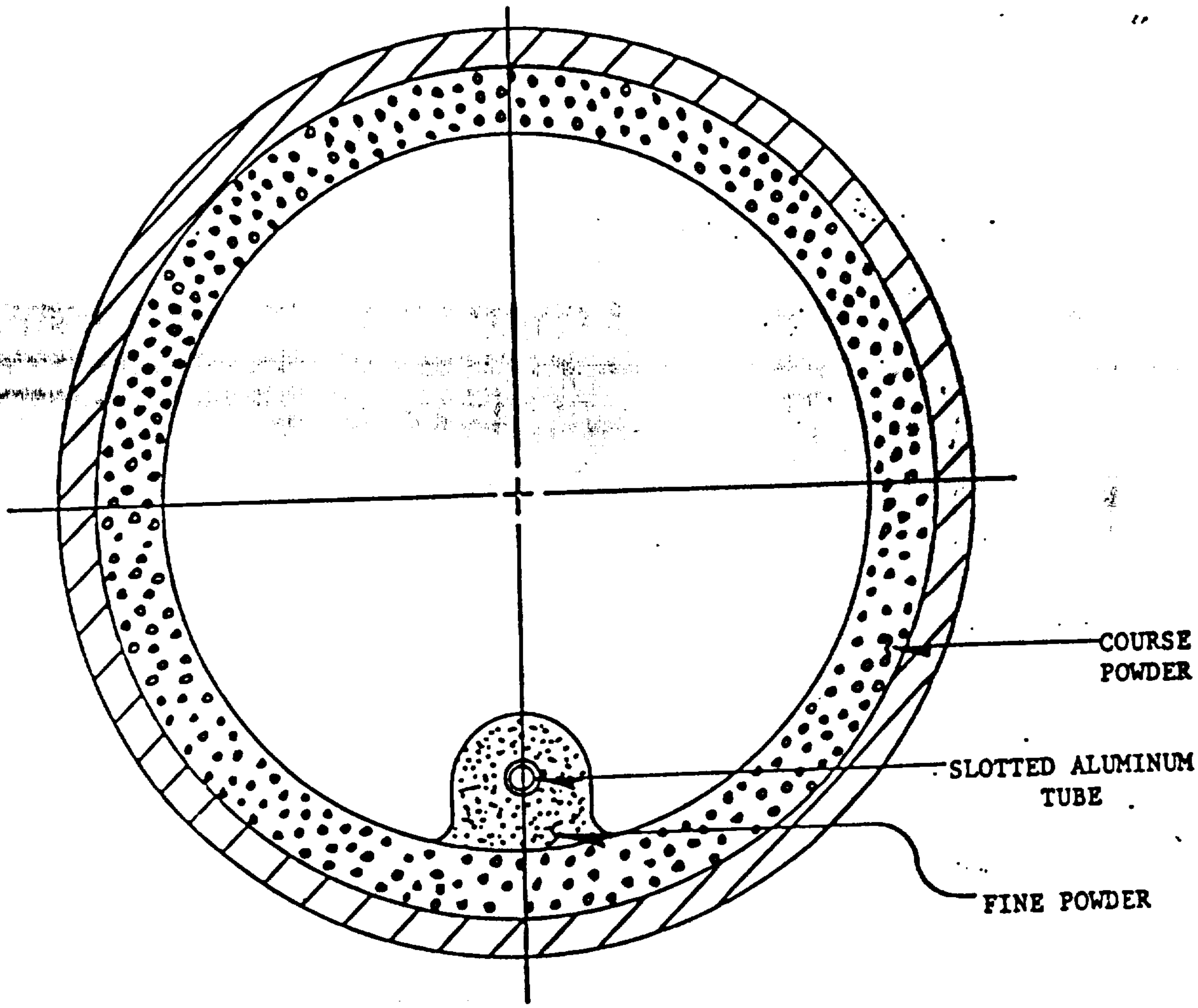


Figure 13. Clamp-on cold plate wick structure

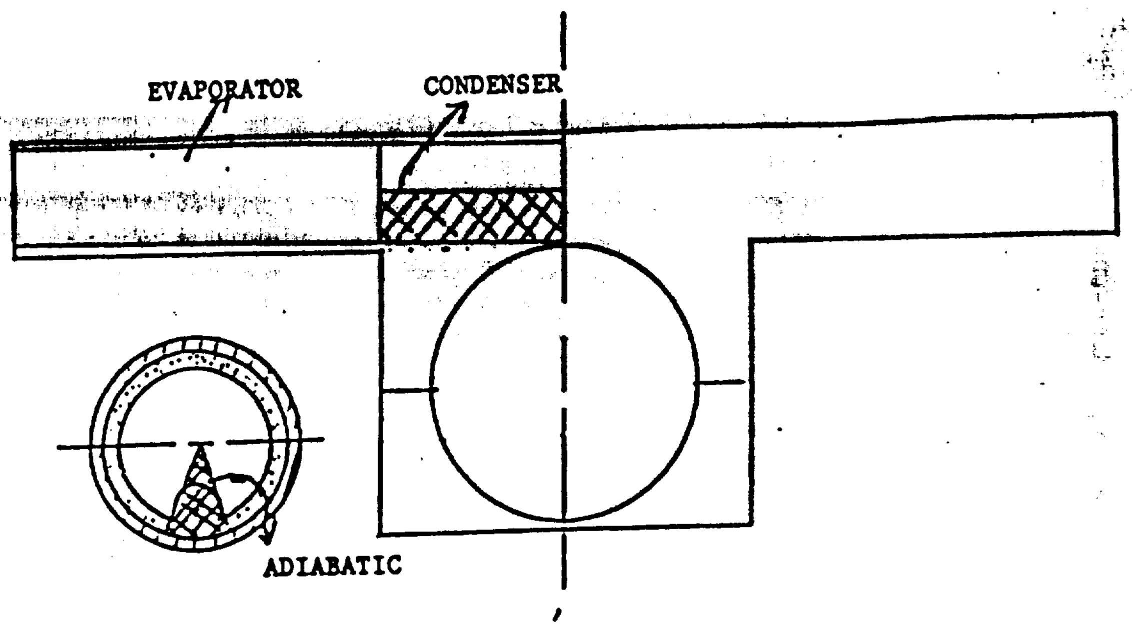


Figure 14. Evaporator adiabatic and condenser areas used to model the clamp-on cold plates

the clamp plus the upper half of the heat pipe above the clamp. The condenser area was assumed to be equal to the bottom half of the heat pipe minus the area of the artery in the vicinity of the clamp. Table 4 is a listing of the predicted thermal performance.

TABLE 4. Predicted Cold Plate Performance

WORKING FLUID	PLATE SIZE (CM)	*CALCULATED POWER TRANSFERRED (W)	CALCULATED ULTIMATE POWER TRANSFERRED (W)
Acetone	30 x 30	1134	2017
Acetone	30 x 60	1288	2169
Acetone	30 x 90	1316	2214
Ammonia	30 x 30	1242	6957
Ammonia	30 x 60	1345	7484
Ammonia	30 x 90	1368	7648

*Assumes a 5°C heat pipe delta-T is equivalent to a 10°C overall delta-T.

The predicted powers are lower than the 2000 - 3000 W goal, however it was mutually (Thermacore/NASA) decided that verifying the vapor resistant artery concepts was more important than the arbitrarily (state-of-the-art) set goal.

4.3 FABRICATION

The cold plates were fabricated from a solid block of 1100 aluminum. This was done for two reasons. First, the integral plate/clamp eliminates a particularly large interface delta-T and secondly, the integral heat pipes with a sintered in place wick structure eliminates a potentially large interface delta-T.

As mentioned previously, the wick structure, sintered aluminum powder, was sintered in place in a vacuum environment. Process controls and wick samples were used to achieve the desired wick pore radius, permeability properties. The sintered wick structure was accomplished using a two-step procedure. The first step was to sinter the coarse powder circumferential wick; and, the second step was to sinter the fine powder/artery liner arterial wick.

All end caps and fill tubes were gas tungsten arc welded (GTAW) using 1100 aluminum as a filler metal. Nupro SS-4H stainless steel bellows valves were attached to each of the fill tubes for ease of filling and venting during the test program. Each of the plates was operated with ammonia for more than 1000 hours at 40°C and then vented to remove any non-condensible gas generated from contaminants after the sintering process. Following this cleaning process, the fill tubes were pinched off and GTAW to seal.

4.4 TEST MATRIX

The test matrix for the clamp-on cold plates was selected to provide the following:

- Performance as a function of plate dimensions
- Performance as a function of working fluid
- Performance as a function of plate orientation (tilt against gravity)
- Performance as a function of heat input area

Three clamp-on cold plates were fabricated (30 x 30 cm, 30 x 60 cm, and 30 x 90 cm) and tested with both acetone and ammonia at three different plate orientations (horizontal, 2 inches, and 4 inches against gravity) with four different heater configurations. In all, 72 tests were run, the complete test matrix is shown in Table 5.

TABLE 5. Clamp-On Cold Plate Test Matrix

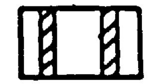
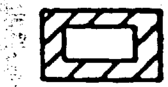
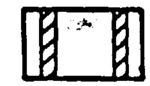
Plate Dimensions (cm)	Working Fluid	Plate Orientation (Inches of Tilt)	Heated Area			
						
30 x 30	Acetone	0" (horizontal)	"	"	"	"
30 x 30	Acetone	2"	"	"	"	"
30 x 30	Acetone	4"	"	"	"	"
30 x 30	Ammonia	0" (horizontal)	"	"	"	"
30 x 30	Ammonia	2"	"	"	"	"
30 x 30	Ammonia	4"	"	"	"	"
30 x 60	Acetone	0" (horizontal)	"	"	"	"
30 x 60	Acetone	2"	"	"	"	"
30 x 60	Acetone	4"	"	"	"	"

TABLE 5. Clamp-On Cold Plate Test Matrix (continued)

Plate Dimensions (cm)	Working Fluid	Plate Orientation (Inches of Tilt)	Heated Area			
30 x 60	Ammonia	0" (horizontal)				
30 x 60	Ammonia	2"	"	"	"	"
30 x 60	Ammonia	4"	"	"	"	"
30 x 90	Acetone	0" (horizontal)	"	"	"	"
30 x 90	Acetone	2"	"	"	"	"
30 x 90	Acetone	4"	"	"	"	"
30 x 90	Ammonia	0" (horizontal)	"	"	"	"
30 x 90	Ammonia	2"	"	"	"	"
30 x 90	Ammonia	4"	"	"	"	"

4.5 TEST PROCEDURE

The following is a step by step test procedure for the clamp-on cold plates. Figure 15 is a sketch of the test set-up and Figure 16 is a photograph of the actual test set-up.

A. Process the heat pipe with either acetone or ammonia

B. Attach the clamp-on cold plate to the H_2O cooled simulated thermal bus using .002" thick indium foil as the interface material. Torque each of the 12 socket head cap screws to 14 ft-lbs.

C. Orient the plate at either 0 inch, 2 inch, or 4 inch tilt pivoted about the thermal bus axis. (See Figure 13).

D. Apply Minco thin foil heaters to the heat input surface as shown in the test matrix.

E. Apply approximately two-inches of Kaowool insulation over the heater input area and clamp in place.

F. Turn on the water to the simulated H_2O cooled thermal bus at approximately 100 ml/min. Temperature and flow rate should be such that the maximum plate temperature is less than 50°C.

G. Energize the heaters and leave undisturbed to reach steady state.

H. When steady state is reached, record all temperatures, flow rate, and heater power.

I. Increase heat load and repeat Steps G to I until the delta-T between the edge of the plate and the thermal bus exceeds 10°C.

4.6 RESULTS AND CONCLUSIONS

A total of 72 tests were run to complete the test matrix. An example of the test data is shown in Figure 16. The plot shows the temperature difference between outer edge of the plate and the interface and thermal bus as shown in Figure 14, versus the total power transferred. For any given test, the temperature difference between the interface value and the thermal bus value is the delta-T of conduction across the interface gap. Appendix A shows the plots for each of the 72 tests.

Table 6 is a comparison of calculated and actual performance for the plates operating at the 10°C delta-T limit. The calculated performance based on evaporator, adiabatic, and condenser areas are shown in Figure 17.

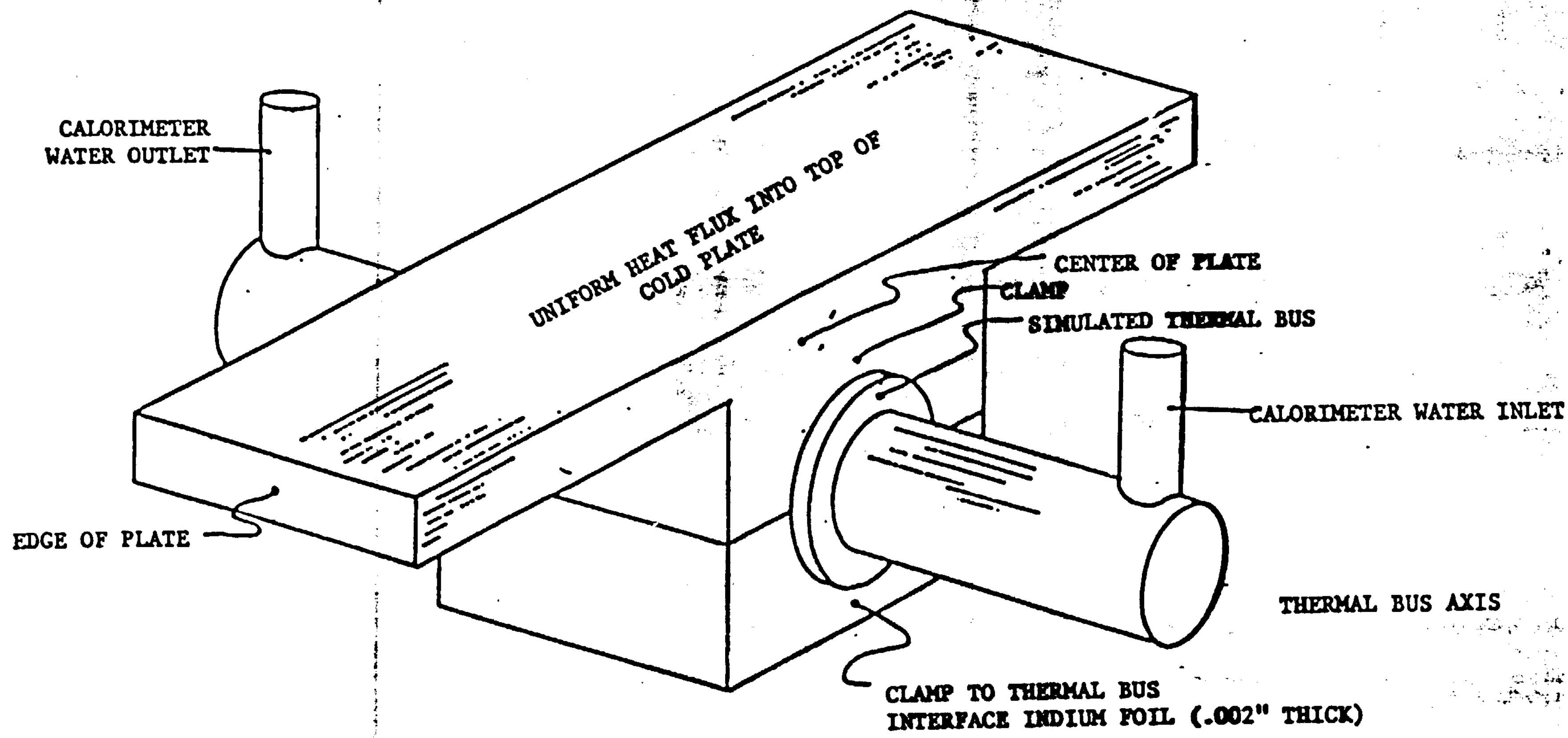


Figure 15. Cold plate test set-up

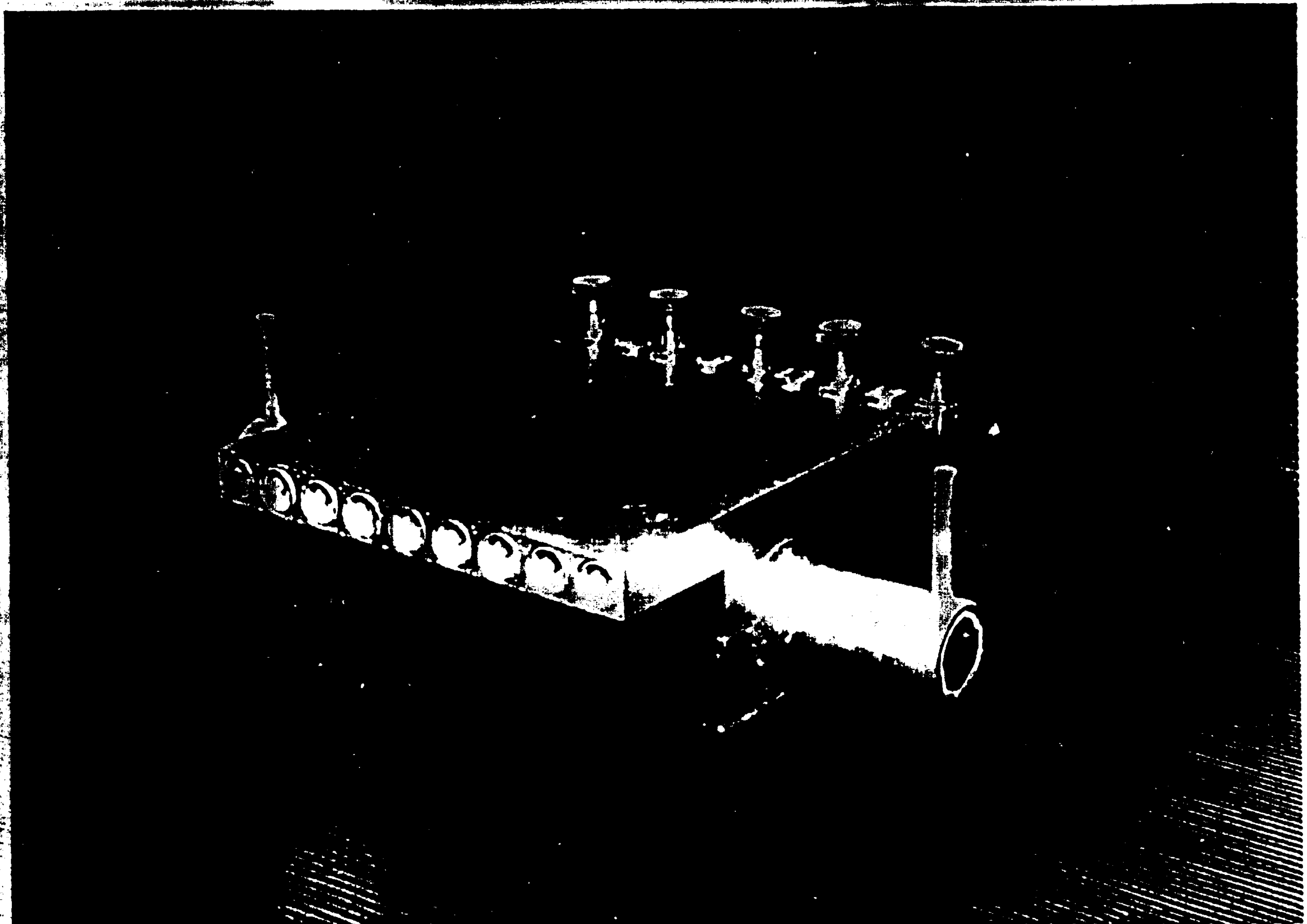
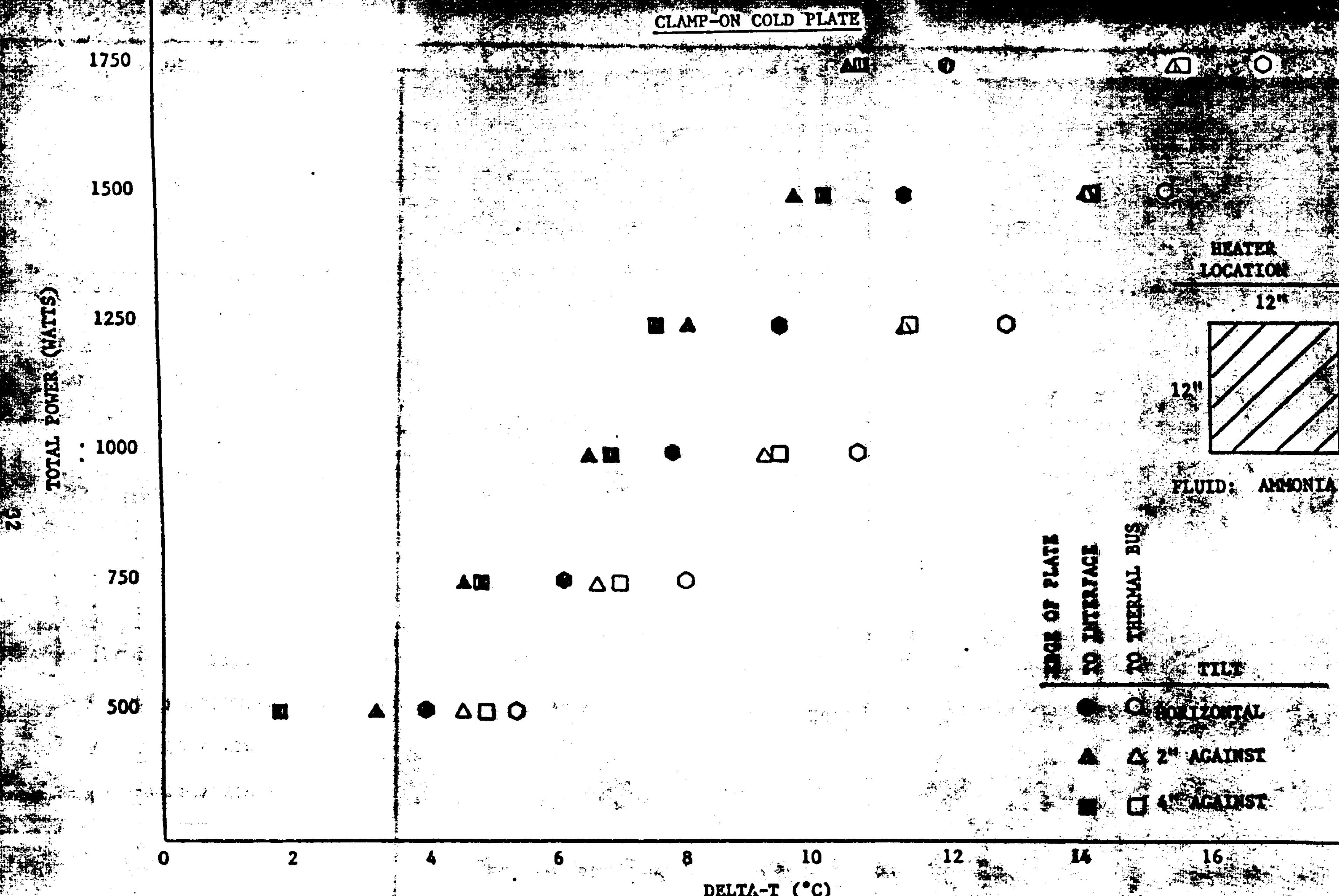


Figure 16.— Photograph of the actual test setup.



PERFORMANCE VERSUS DELTA-T FOR SEVERAL TILT ANGLES

Figure 17. Example of test data

The condenser area is assumed to equal the bottom half of each pipe minus the area of the artery in the vicinity of the thermal bus. This assumption is verified by the test data shown in Table 6.

TABLE 6. Comparison of Calculated and Actual Cold Plate Performance

WORKING FLUID	DIMENSIONS (CM)	*CALCULATED POWER (W)	ACTUAL POWER (W)
Acetone	30 x 30	1134	900
Acetone	30 x 60	1288	875
Acetone	30 x 90	1316	1125
Ammonia	30 x 30	1242	1000
Ammonia	30 x 60	1345	1250
Ammonia	30 x 90	1368	1500

*Assumes a 5°C heat pipe delta-T is equivalent to a 10°C overall delta-T.

Both the predicted performance results and the actual test performance indicate the two most important design limitations. The first is the delta-T of conduction across the clamp and interface between the clamp and the thermal bus. Of the 10°C total allowable delta-T from the outer edge of the plate to the thermal bus surface, 5°C is necessary for the delta-T of conduction mentioned above. The second design limitation is the delta-T of condensation inside of the heat pipe. This delta-T is 70% of the total heat pipe delta-T. Therefore, in order to increase the performance of these cold plates, a clamping mechanism would have to be designed to include a larger heat pipe condensation area, and the clamp itself must have a lower thermal resistance.

TABLE 7. Calculated Maximum Power (Capillary Limited)*

WORKING FLUID	TILT	POWER (W) DELTA T (°C) 30 X 30 cm	30 X 60 CM	30 X 90 CM
Ammonia	Horizontal	6957 (27)	7848 (27)	7648 (27)
	2" Tilt	5953 (23)	6415 (23)	6554 (23)
	4" tilt	4963 (19)	5353 (20)	5464 (20)
Acetone	Horizontal	2017 (7)	2169 (8)	2214 (8)
	2" tilt	1662 (7)	1791 (7)	1829 (7)
	4" tilt	1312 (6)	1417 (6)	1445 (6)

*Assume artery leak tightness equivalent to sintered wick capability and vapor does not penetrate into the artery.

Table 7 is a listing of the capillary limited maximum power that could be expected with the present design. It is clear that the plates utilizing ammonia as the working fluid have a potential to carry substantial power. However, the total delta-T's (from plate outer edge to thermal bus) using the present clamping mechanism are quite large (19-27°C).

After reviewing the performance data and predictions, the following conclusions and recommendations can be reached.

- The 10°C delta-T limit allows only a small increase in power carrying capability with a substantial increase in plate size. This indicates that the performance is limited by the heat pipe delta-T of condensation and the delta-T of conduction through the clamping mechanism and interface gap.

- A uniformly distributed heat load is the best loading for the present design. However, strip loads are also easily accommodated provided the loads are perpendicular to the heat pipes.

- The 30 x 30 cm and 30 x 60 cm plates performed as expected with acetone and ammonia. The 30 x 90 cm plate performed as expected with acetone, however, it appears that it was not maintaining primed arteries with ammonia.

- Higher power carrying capabilities are easily accommodated by the present heat pipe design, however, a larger condenser area and a lower thermal resistance clamp and interface are required to meet the 10°C overall delta-T limit.

5.0 FLEXIBLE HEAT PIPE COLD PLATE AND THERMAL BUS COUPLER

The development of the flexible heat pipe cold plate (FHPCP) was conducted following the identification of the design requirements shown in Section 5.1.

5.1 REQUIREMENTS

The design requirements for the FHPCP are shown in Table 8.

TABLE 8. Flexible Heat Pipe Cold Plate Design Requirements

PARAMETER	MAGNITUDE
Material	1100 Aluminum
Working Fluid	Ammonia, Acetone
Mass	Minimize
Operating Temperature (°C)	0-40
Dimensions (cm)	30 x 30, 10 x 2.54 diameter through holes
	30 x 30, 16 x 1.58 diameter through holes
Temperature (°C)	8 -10
Total Power (W)	>1000

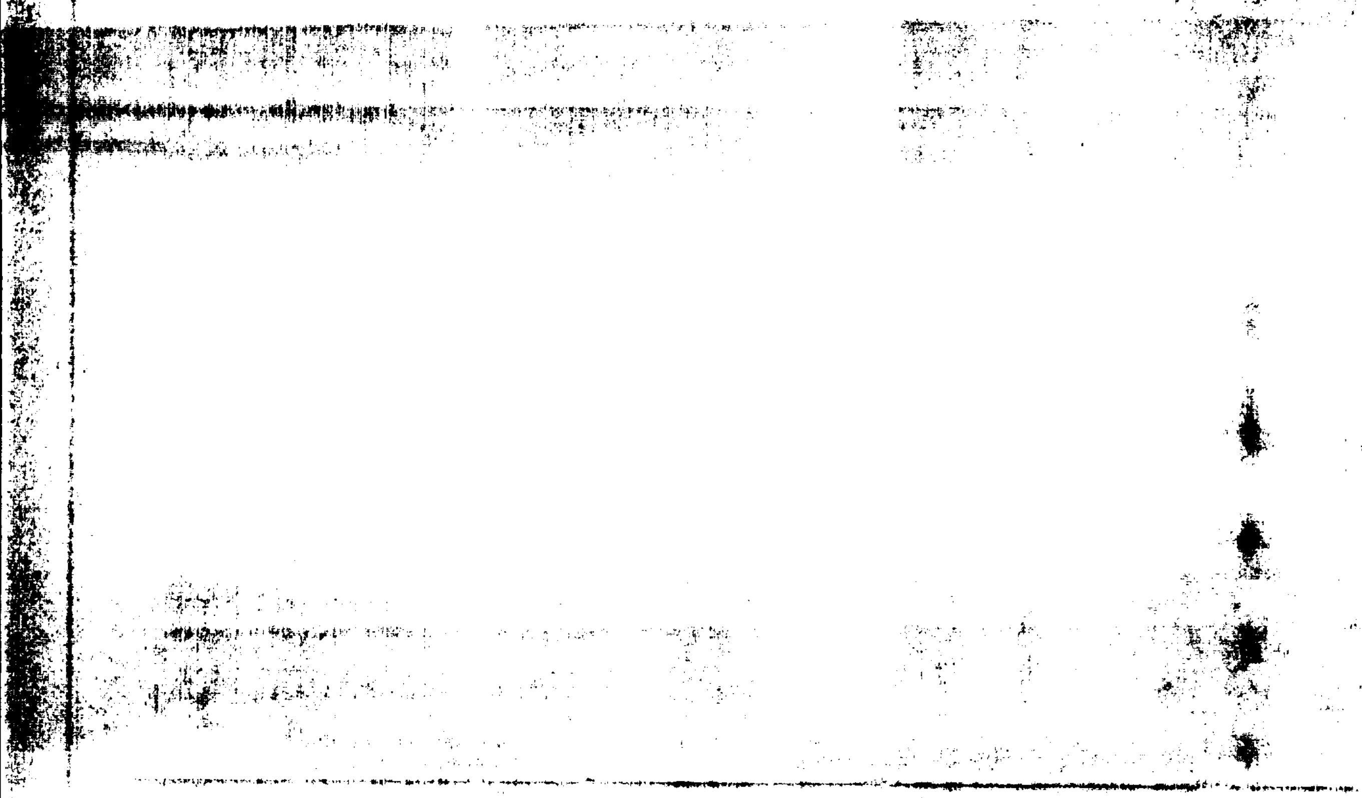
5.2 FLEXIBLE HEAT PIPE COLD PLATE DESIGN

The flexible heat pipe cold plate shown in Figure 18 was designed using the requirements in Table 8 as design goals. The design of the FHPCP began with an evaluation of alternative wick structures shown in Figure 19. The wick structures were evaluated in terms of anticipated performance, ease and cost of fabrication, weight, artery priming, tolerance to boiling and overall heat pipe Delta-T.

Based on fabrication considerations and weight, the round geometry, design F (Figure 19), was selected as the primary design. Pedestal artery concepts in both a 2.54 cm diameter and a 1.58 cm diameter were selected as the most promising wick structure designs.

All of the wick structure concepts in Figure 19 are tunnel artery designs. Tunnel artery designs are proven high performance wick structures, but tunnel arteries do not inhibit vapor penetration into the

18 Flexible heat pipe cold plate



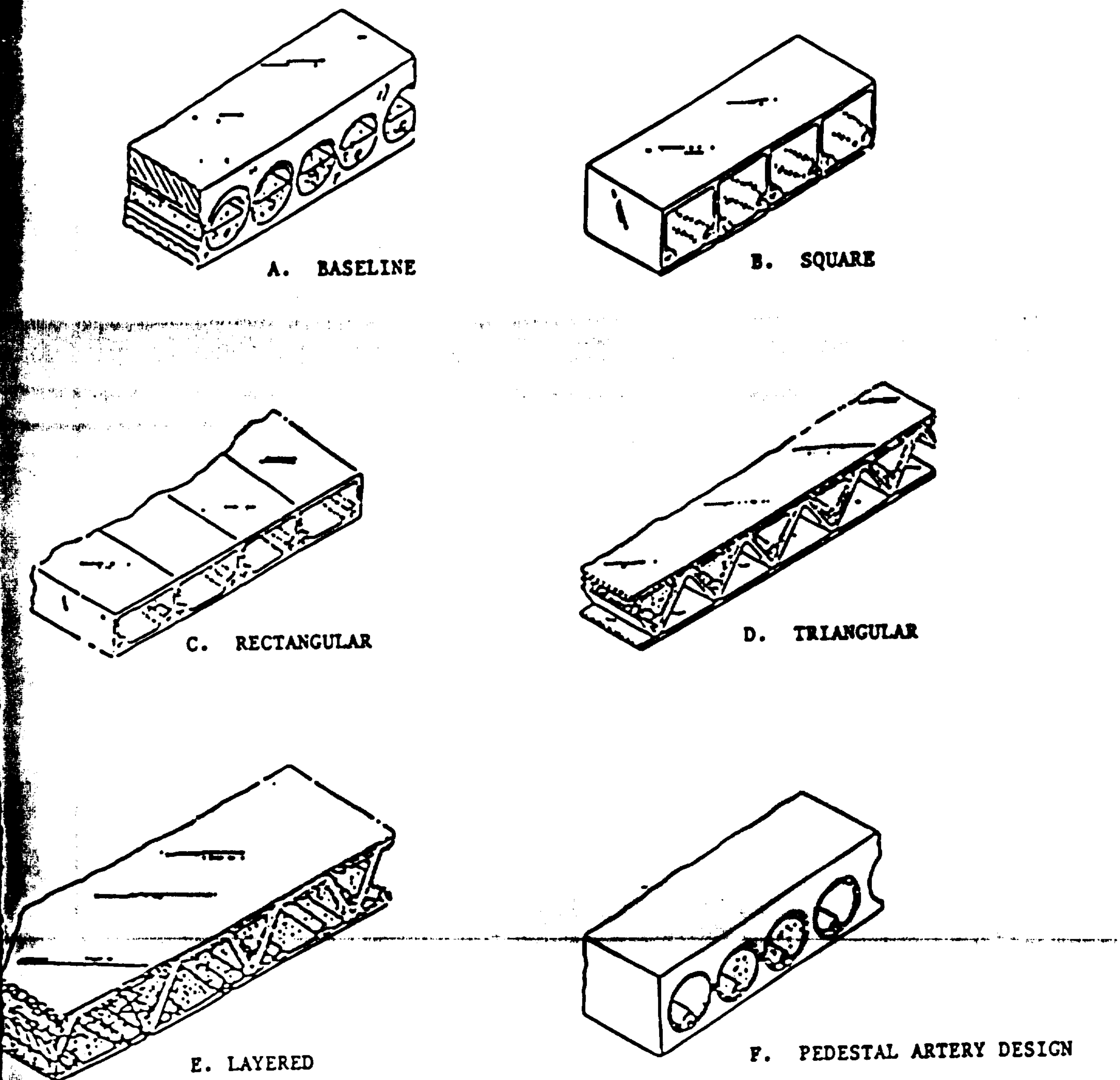


Figure 19. Alternative cold plate wick structure design

artery. Vapor generated at the wall/wick interface penetrates the artery and causes it to deprime. When this happens, the heat pipe can no longer transfer power and dries out. To prevent vapor from penetrating the artery, a wick structure was designed with a surface tension resistance to the vapor space less then the surface tension resistance to the artery. Figure 18 shows the wick and artery design.

The design incorporates a coarse powder circumferential wick and a fine powder artery with a slotted artery liner. The coarse powder has excellent permeability and pore radius properties and is responsible for the heat pipe's high performance capability. The fine powder and slotted tube impose a high surface tension barrier around the artery to prevent vapor from penetrating and depriming the artery.

Figure 20 shows the artery features incorporated in this design to assure tolerance to boiling. The remainder of our effort centered around the heat pipe delta-T, weight and erase and cost of fabrication.

The square, rectangular, layered and triangular designs would require thick walls to avoid buckling due to the internal vapor pressure of ammonia. This would also increase the Delta-T through the cold plate wall. Fabrication of these designs would be difficult implying a high cost of manufacturing.

The circular geometry cold plate designs are lightweight, cost effective and provide a low delta-T. Table 9 describes the circular, square, and triangular concepts in terms of dimension, number of heat pipes per cold plate, weight and delta-T through the cold plate wall.

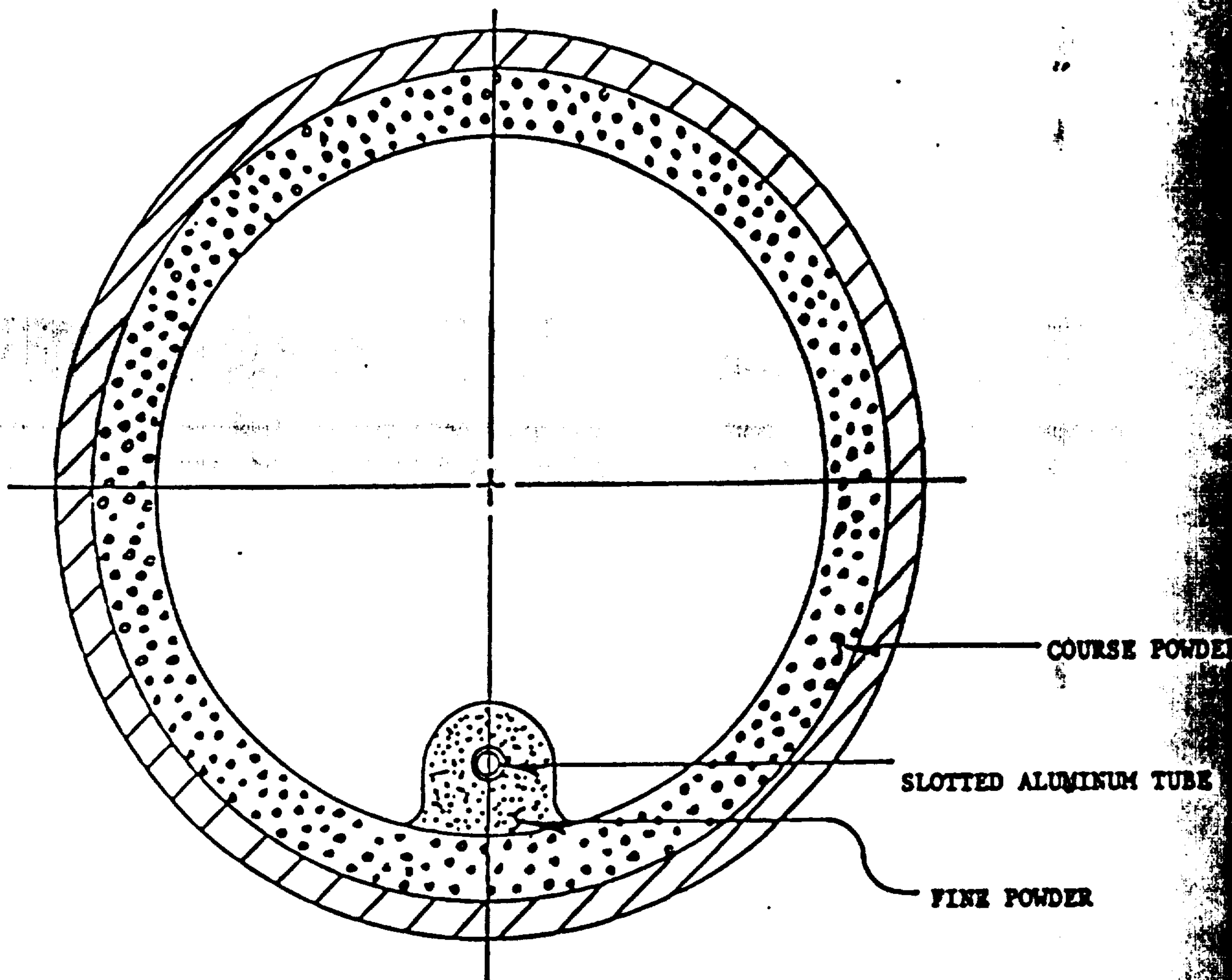


Figure 20. Vapor penetration resistant pedestal artery design

TABLE 9. Wick Structure Analysis

WICK GEOMETRY	HEAT PIPE DIMENSIONS	# HEAT PIPES/ COLD PLATE	WEIGHT Kg	DELTA-T COLD PLATE (°C)
Circular	2.54 cm dia.	10	4.68	0.04
Square	2.54 cm x 2.54 cm	9	6.08	0.42
Triangular	2.54 cm x 2.54 cm x 2.54 cm	14	7.09	0.27
Circular	1.588 cm	16	3.39	0.03
Square	1.588 cm x 1.588 cm	15	4.14	0.22
Triangular	1.588 cm x 1.588 cm x 1.588 cm	24	4.85	0.14

Based on the thermal and geometric requirements established in Task 1, the design parameters for the two circular geometry FHPCP's were established. Details of the FHPCP geometry and wick structure are listed in Table 10.

5.3 FABRICATION, FLEXIBLE HEAT PIPE COLD PLATE

Based on the established thermal designs for the 2.54 cm diameter and the 1.58 cm diameter FHPCP concepts, each unit was built as outlined below.

- Sinter coarse circumferential powder metal wick. All 10 or 16 heat pipes were sintered simultaneously.
- Insert slotted artery tubes and sinter fine powder metal wick. Again, all 10 to 16 heat pipes were sintered simultaneously.
- Sinter condenser wick structure as outlined above
- Test the arteries for the amount of pressure they can hold. A pressure of 8.89 cm of water was required when tested in acetone.
- Weld artery manifolds to the 10 or 16 artery tubes.
- Pressure check the arteries again.
- Insert several layers of -325 stainless steel screen into the stainless steel bellows to reduce the possibility of liquid entrainment.
- Weld on manifold end caps to the cold plate.
- Weld Bi-Braze connectors to the manifold end cap and condenser. Bi-Braze connectors are stainless steel to aluminum transition couplings.
- Insert the flexible teflon shrink tubing artery into the stainless steel bellows and seal to the cold plate manifold and condenser arteries.
- Weld the assembly together including the condenser end cap.
- Check for leaks with a helium mass spectrometer.

TABLE 10. Flexible Heat Pipe Cold Plate Design Parameters

COLD PLATE:	30 cm x 30 cm, 1110 aluminum Concept I: 10-2.54 cm diameter heat pipes Concept II: 16-1.58 cm diameter heat pipes
CONDENSER:	Length - 30 cm, 1100 aluminum
ADIABATIC:	Length - 76.2 cm, includes stainless steel flexible bellows
WICK:	Evaporator and Condenser -60 +150 aluminum powder - circumferential wick -150 +325 aluminum powder - arterial wick (evaporator only)
ARTERY:	0.476 cm diameter artery enclosed in a 0.159 cm powder metal wick Teflon tubing used in flexible bellows
WORKING FLUID:	Ammonia and acetone
PERFORMANCE TESTING:	Powder versus Delta-T Power versus Inclination Against Gravity Distributed Heat Loads

5.4 TEST MATRIX, FLEXIBLE HEAT PIPE - COLD PLATE

A test matrix for the FHPCP is shown in Figure 21. The FHPCP's were performance tested using both ammonia and acetone. Tests were conducted at horizontal, 5.08 cm and 10.16 cm against gravity. The evaporator region was heated using four (4) heater arrangements.

5.5 TEST PROCEDURE, FLEXIBLE HEAT PIPE - COLD PLATE

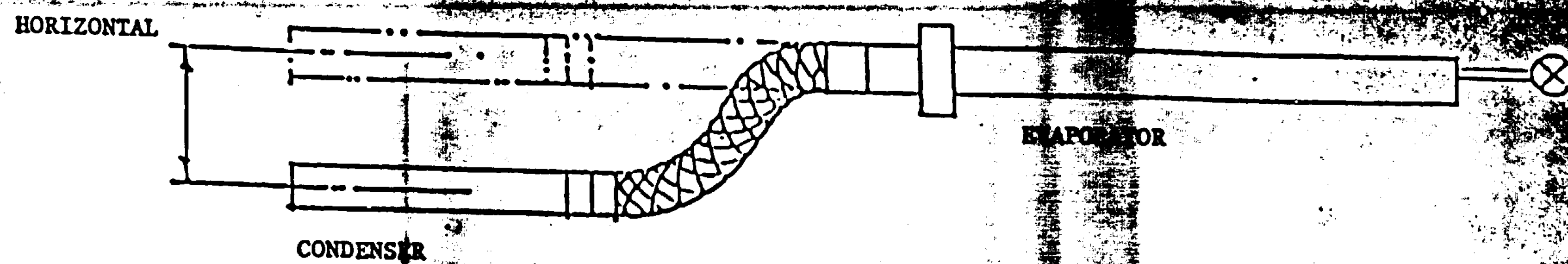
Minco thin foil heaters supplied heat to the FHPCP evaporator in the heater configurations shown in Figure 21. For each heater location, the heat flux was uniform over the heated area. Insulation was placed over the heater and clamped in place. The FHPCP with 2.54 cm diameter holes was processed with acetone and ammonia as the working fluid and oriented at a 0 cm, 5.08 cm and 10.16 cm tilt against gravity, pivoted from the flexible bellows. The FHPCP with 1.58 cm diameter holes was tested with acetone only.

A water cooled heat exchanger was placed around the condenser to simulate the thermal bus. The heaters were turned on and the cold plate was left undisturbed to reach steady state operation. Upon reaching steady state, the temperatures were recorded and the heat load increased. This procedure was repeated until the delta-T between the edge of the cold plate and the condenser exceeded 10°C during testing. A photograph of the FHPCP during performance testing is shown in Figure 22.

5.5.1 Artery Priming

In order for a heat pipe of any design to operate using arteries as the liquid return mechanism, the artery must be "primed." The artery is required to be completely filled with the working fluid with no vapor present. This "priming" mechanism allows the evaporator to draw fluid from the condenser via the artery. If the artery is not "primed" then fluid cannot return to the evaporator and the evaporator wick structure dries out. With the evaporator dried out, the heat pipe can no longer transfer power.

On earth, the artery will prime if the capillary head of the artery is greater than the gravity head of the wick structure. For space the gravity head is eliminated and the artery will self-prime.



HEATED AREA

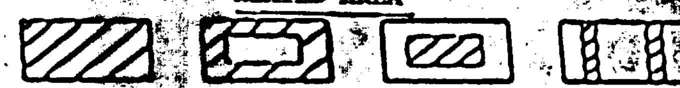
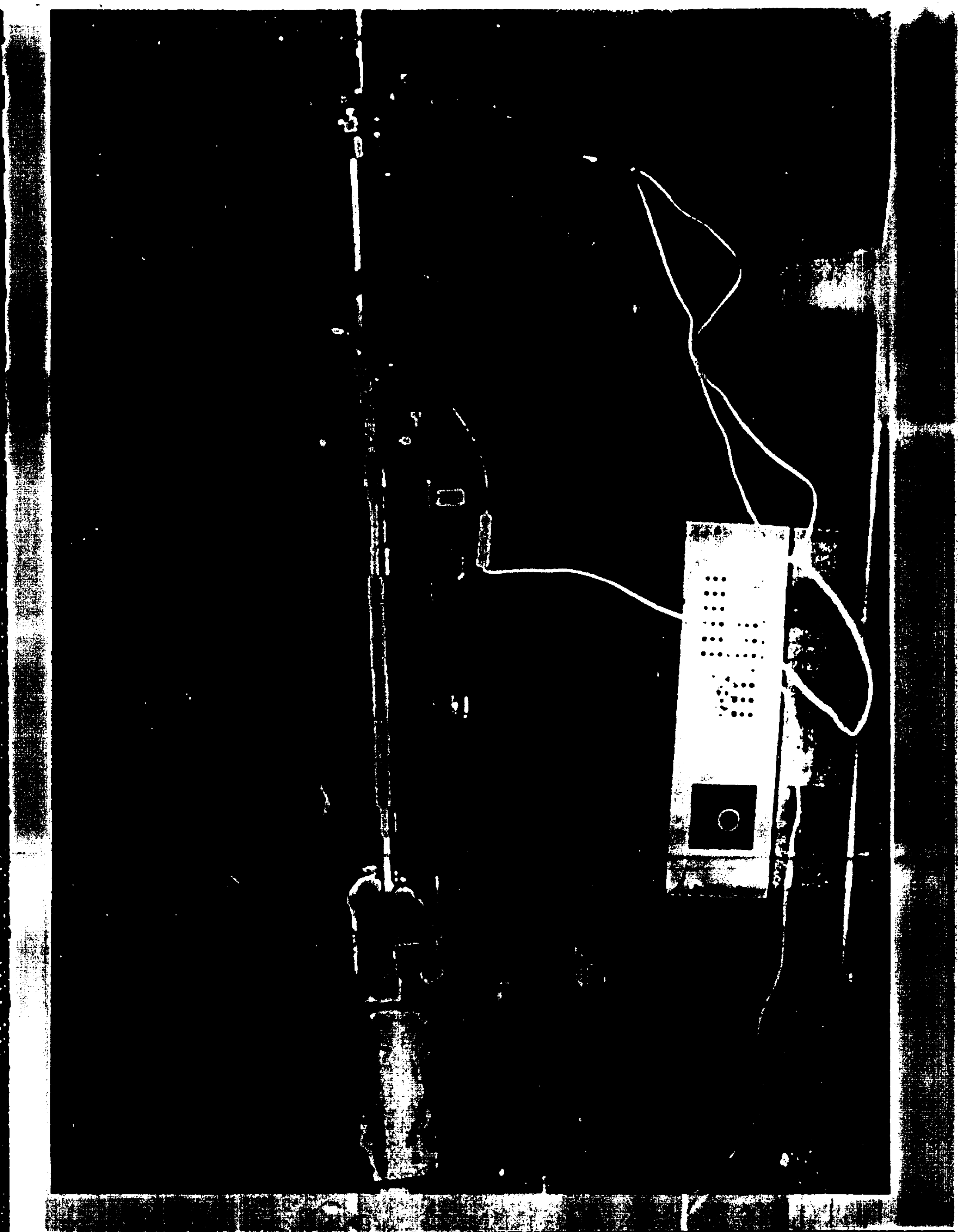


PLATE DIMENSIONS (cm)	WORKING FLUID	PLATE ORIENTATION (INCHES OF TILT)				
30 x 30	Acetone	0" (horizontal)				
30 x 30	Acetone	2"	"	"	"	"
30 x 30	Acetone	4"	"	"	"	"
30 x 30	Ammonia	0" (horizontal)	"	"	"	"
30 x 30	Ammonia	2"	"	"	"	"
30 x 30	Ammonia	4"	"	"	"	"

Figure 21. Flexible heat pipe cold plate test matrix and orientation



The procedure for priming the arteries using acetone as the working fluid was established under NASA Contract Number NAS8-35263, "Heat Transport Across Structural Boundaries." The procedure utilized the vapor pressure of the working fluid to push the liquid into the artery. Any vapor gas present in the artery is condensed by the vapor pressure of the working fluid.

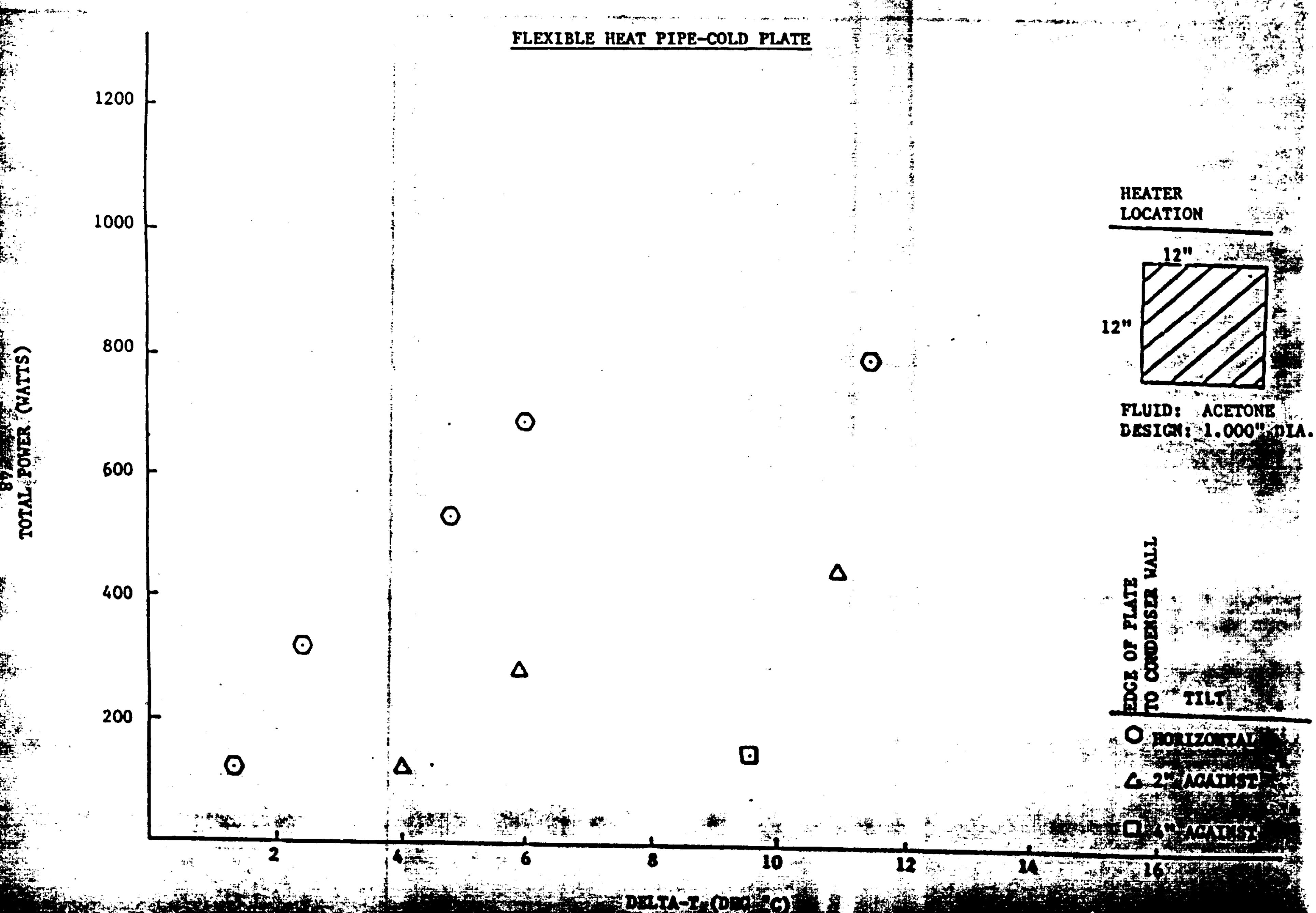
5.6 RESULTS AND CONCLUSION, FLEXIBLE HEAT PIPE COLD PLATE

Results of the performance tests conducted on the 2.54 cm FHPCP design using acetone and ammonia as the working fluid are shown in Figures 23-30. The data plotted shows the delta T between the average temperature of the cold plate evaporator and the condenser wall, as a function of power transferred.

Figures 23-30 show plots of the performance tests using acetone as the working fluid. Tests were conducted at 0.00 cm, 5.08 cm and 10.16 cm inclination against gravity with four different heater geometries on the cold plate surface. Inspection of the performance data indicates that uniform heat loads and strip heat loads normal to the heat pipes performed best.

In Task 1, a maximum delta-T requirement was established at 10°C, with the delta-T being the average temperature of the cold plate evaporator to the temperature of the thermal bus. The test results shown in this section do not include the delta-T of the interface between the FHPCP and the thermal bus receptacle. While conducting performance tests for the FHPCP, a concurrent study was conducted to design and test different interface joint geometries between the plug-in components of the FHPCP and thermal bus receptacle. The purpose of the study was to develop an interface with a low delta-T, thereby increasing the heat transfer capability of the FHPCP while satisfying the 10°C limitation.

Test results indicated that a contact resistance of 1.35 $cm^2 \cdot °C/W$ was achieved for a tapered condenser/thermal bus receptacle joint geometry. Therefore, the FHPCP tested with a uniform heat load heater geometry would yield a maximum heat transfer capability of 800 watts in order to meet the 10°C delta-T requirement. The strip heater geometry would yield a 700 watt maximum heat transfer capability. These heat transfer


FLEXIBLE HEAT PIPE-COLD PLATE
TOTAL POWER (WATTS)
1200
1000
800
600
400
200
2
4
6
8
10
12
14
16
HEATER
LOCATION
12"
12"
FLUID: ACETONE
DESIGN: 1.000" DIA.
EDGE OF PLATE
TO CONDENSER WALL
TILT
HORIZONTAL
2" AGAINST
4" AGAINST

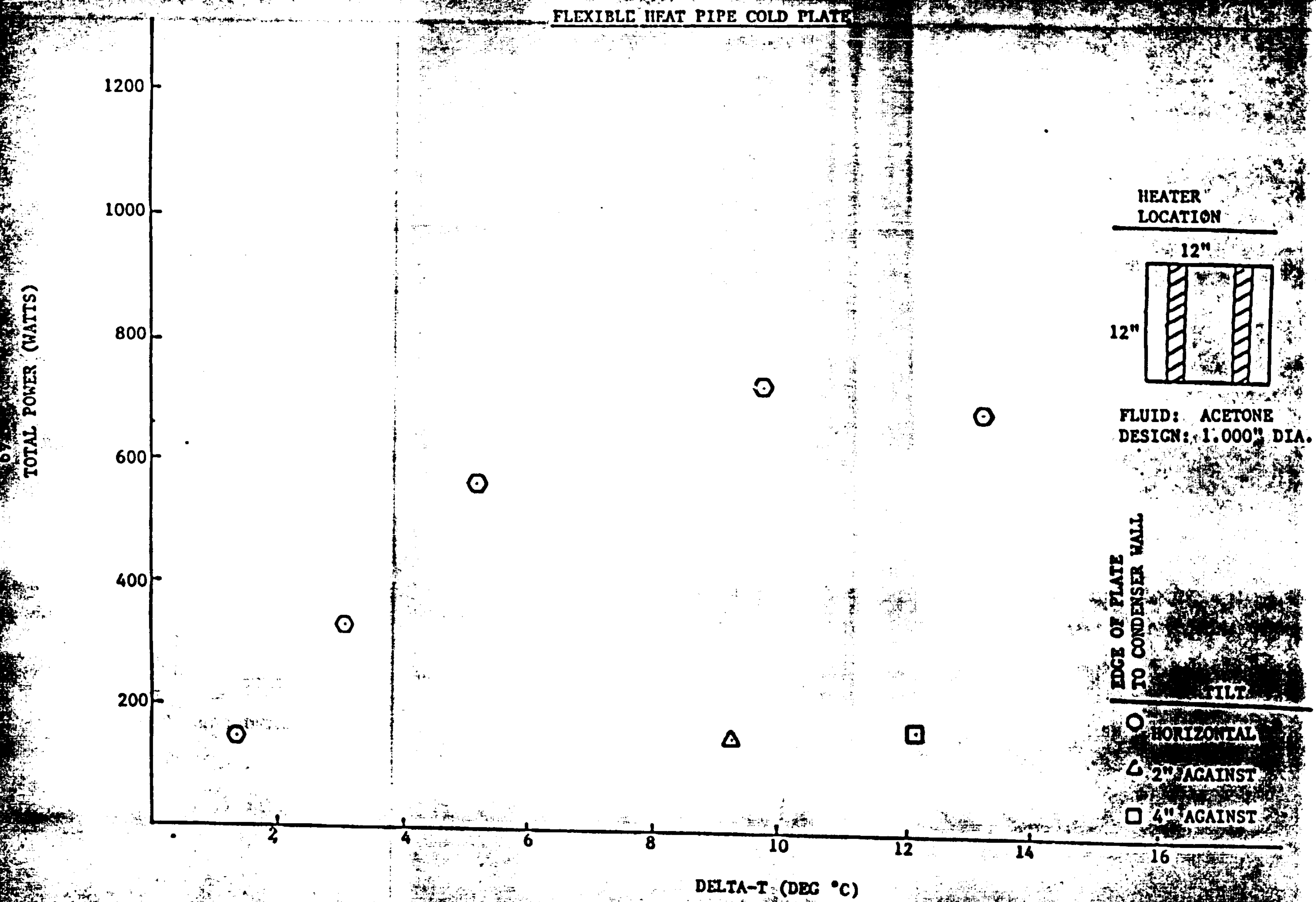


Figure 24. Performance vs. delta-T for several tilt angles

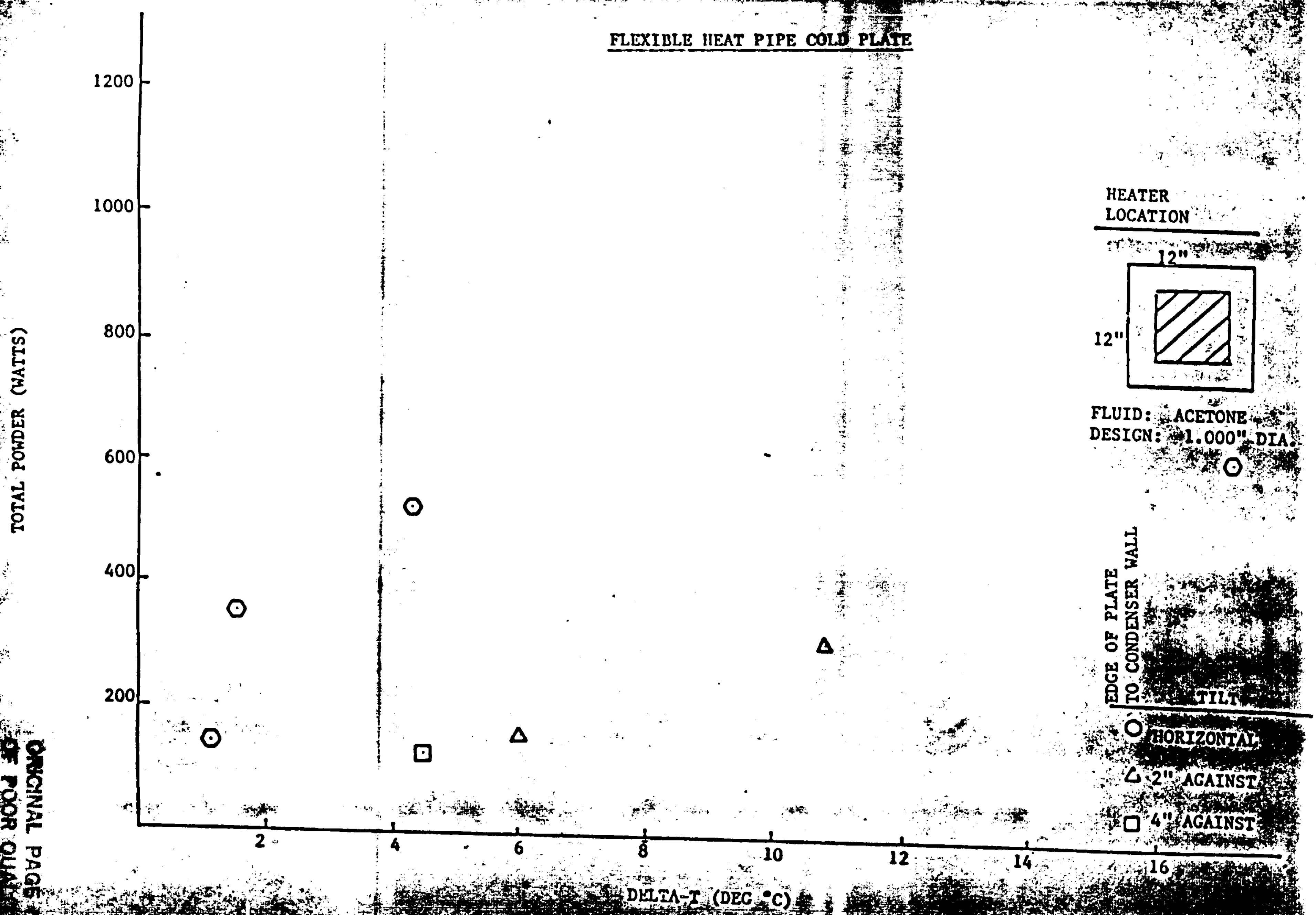


Figure 25. Performance vs. delta-T for several tilt angles

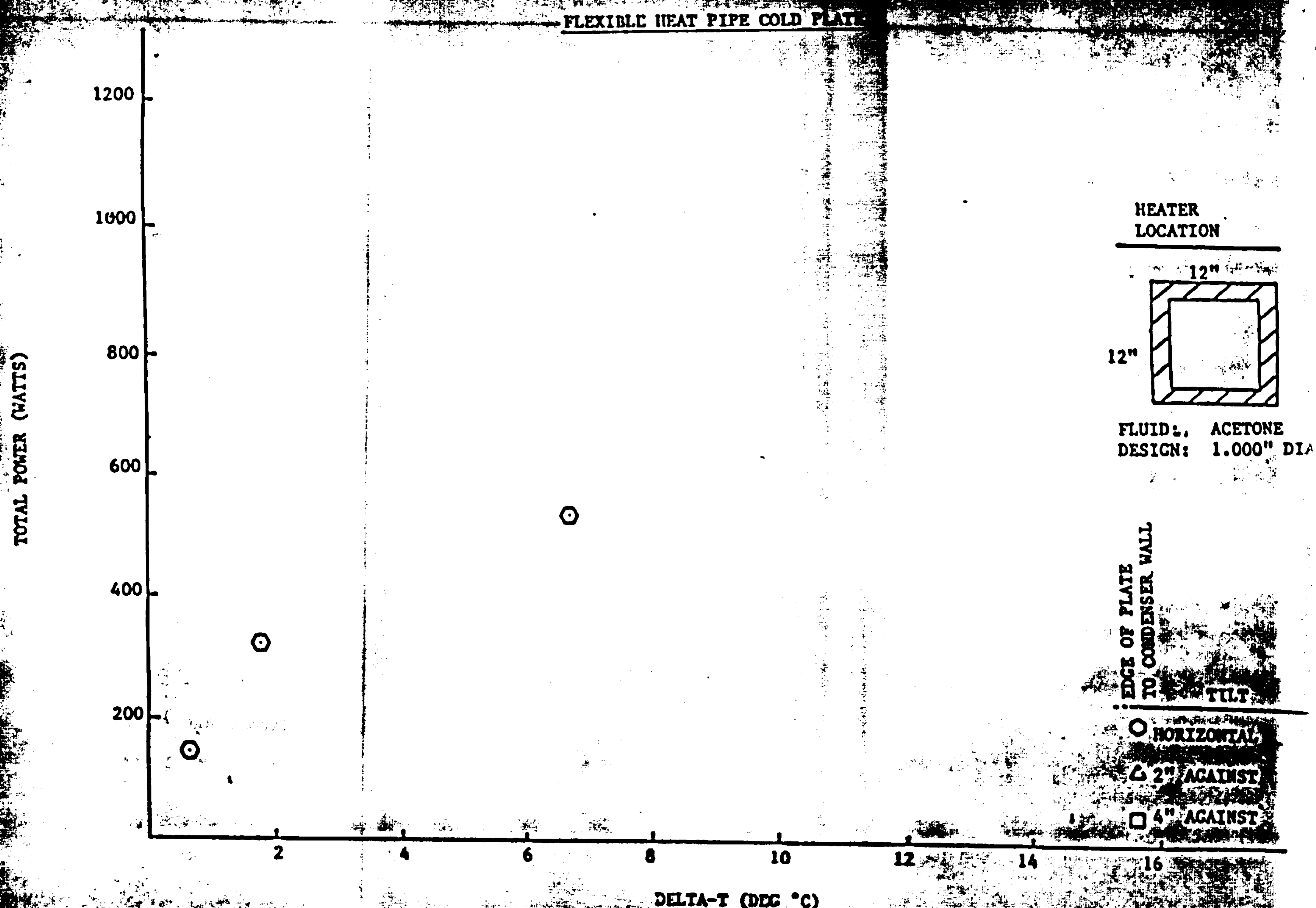


Figure 26. Performance vs. delta-T for several tilt angles

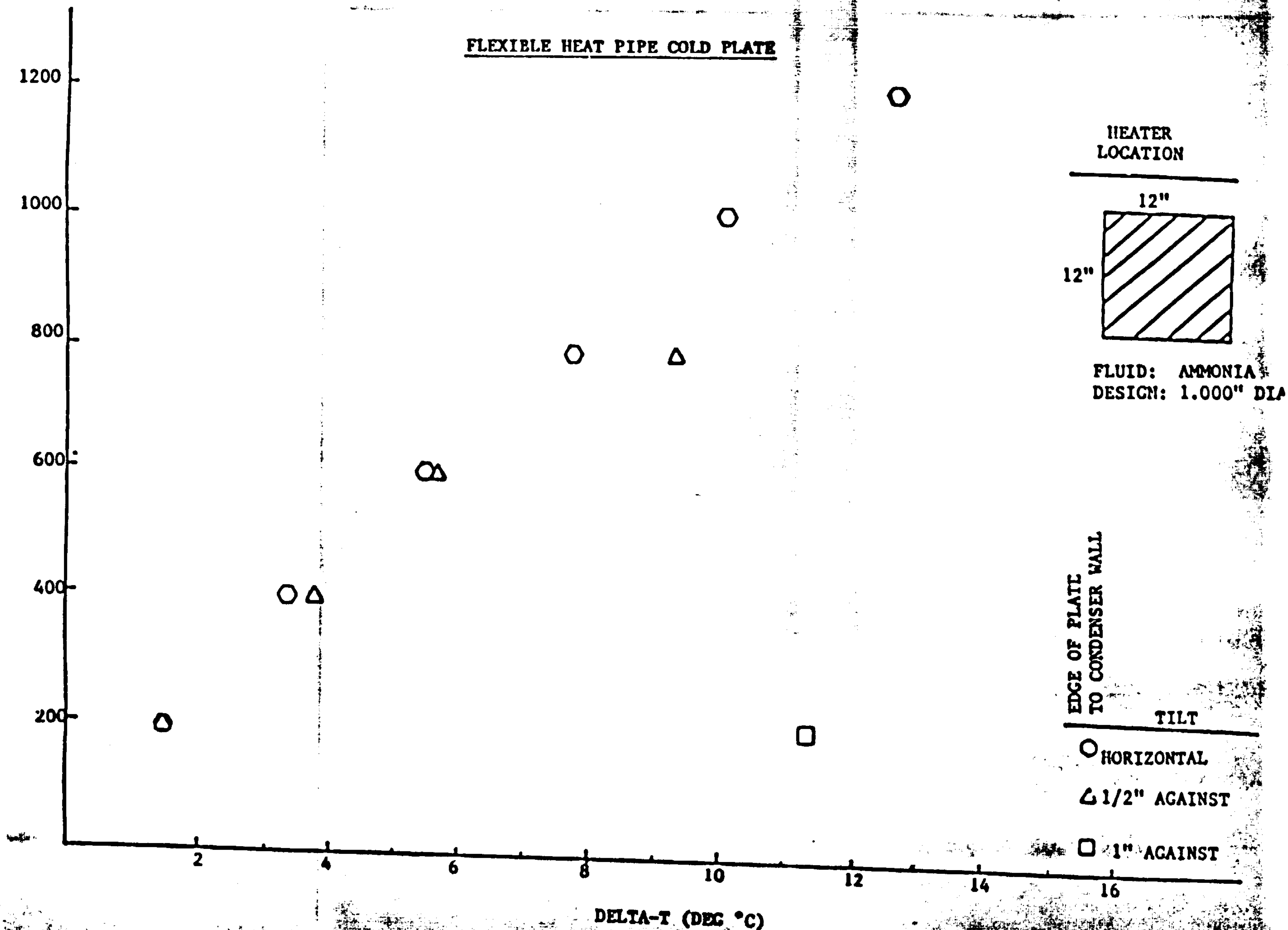


Figure 27. Performance vs. delta-T for several tilt angles

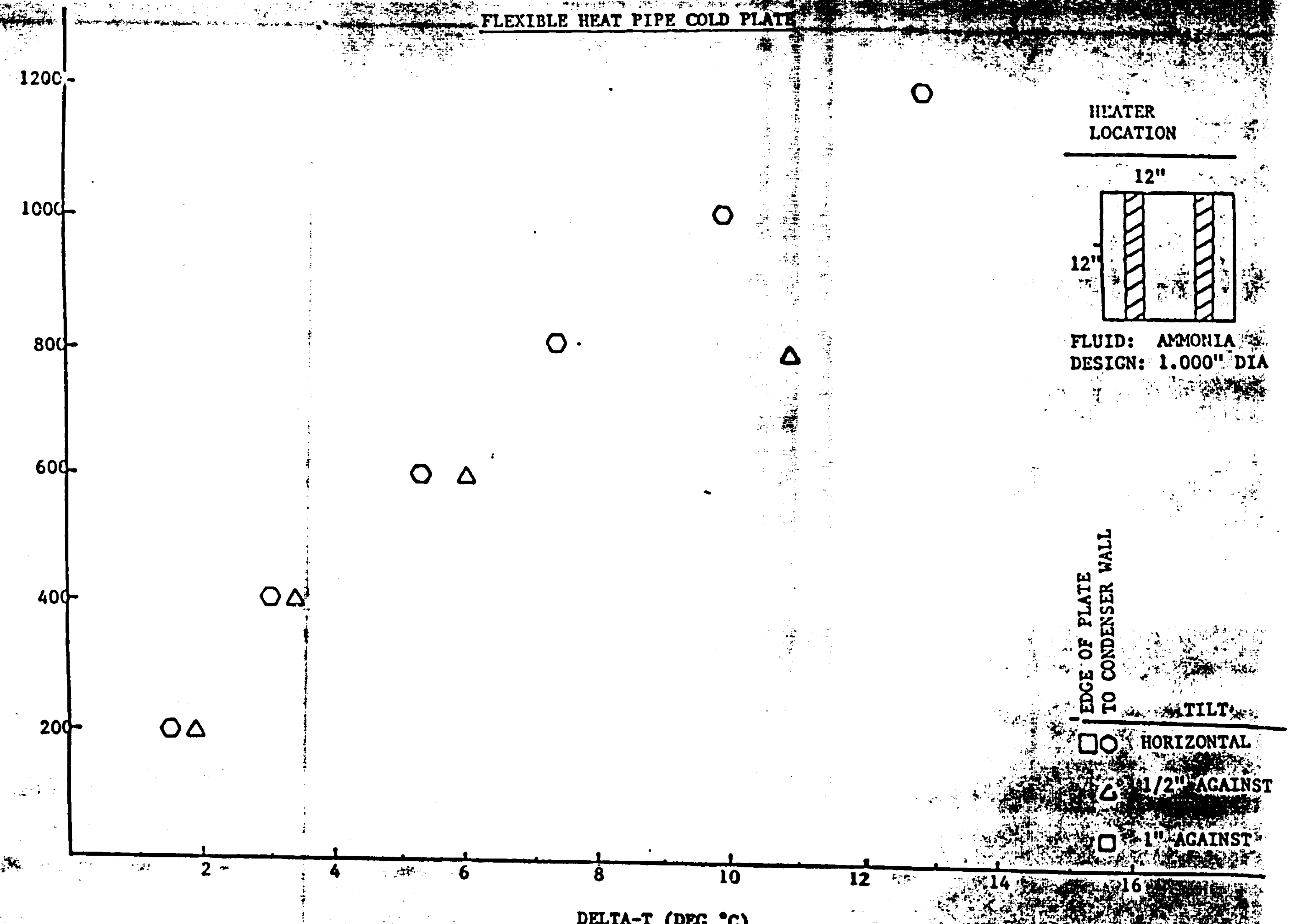


Figure 28. Performance vs. delta-T for several tilt angles

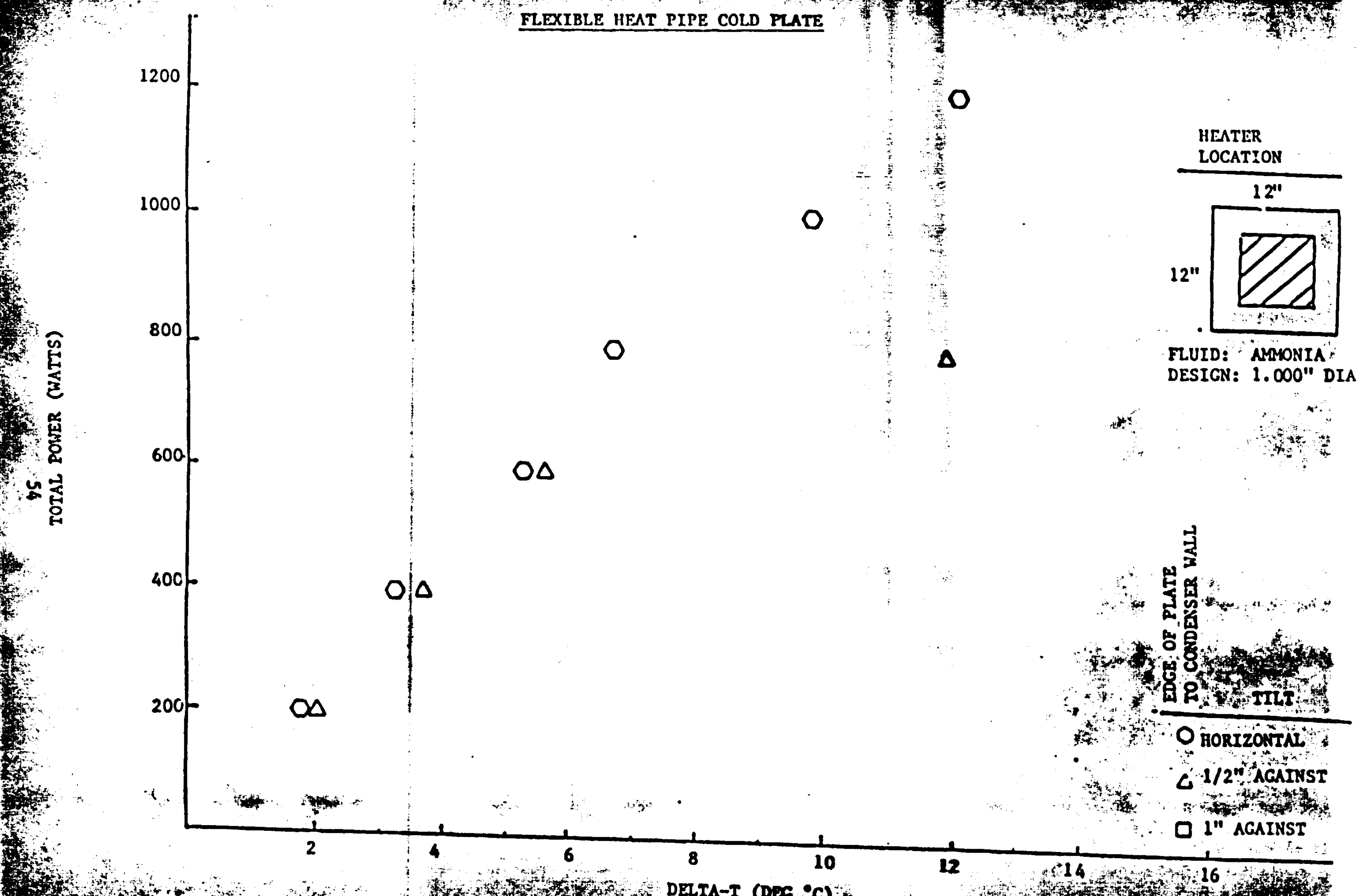


Figure 29. Performance vs. delta-T for several tilt angles

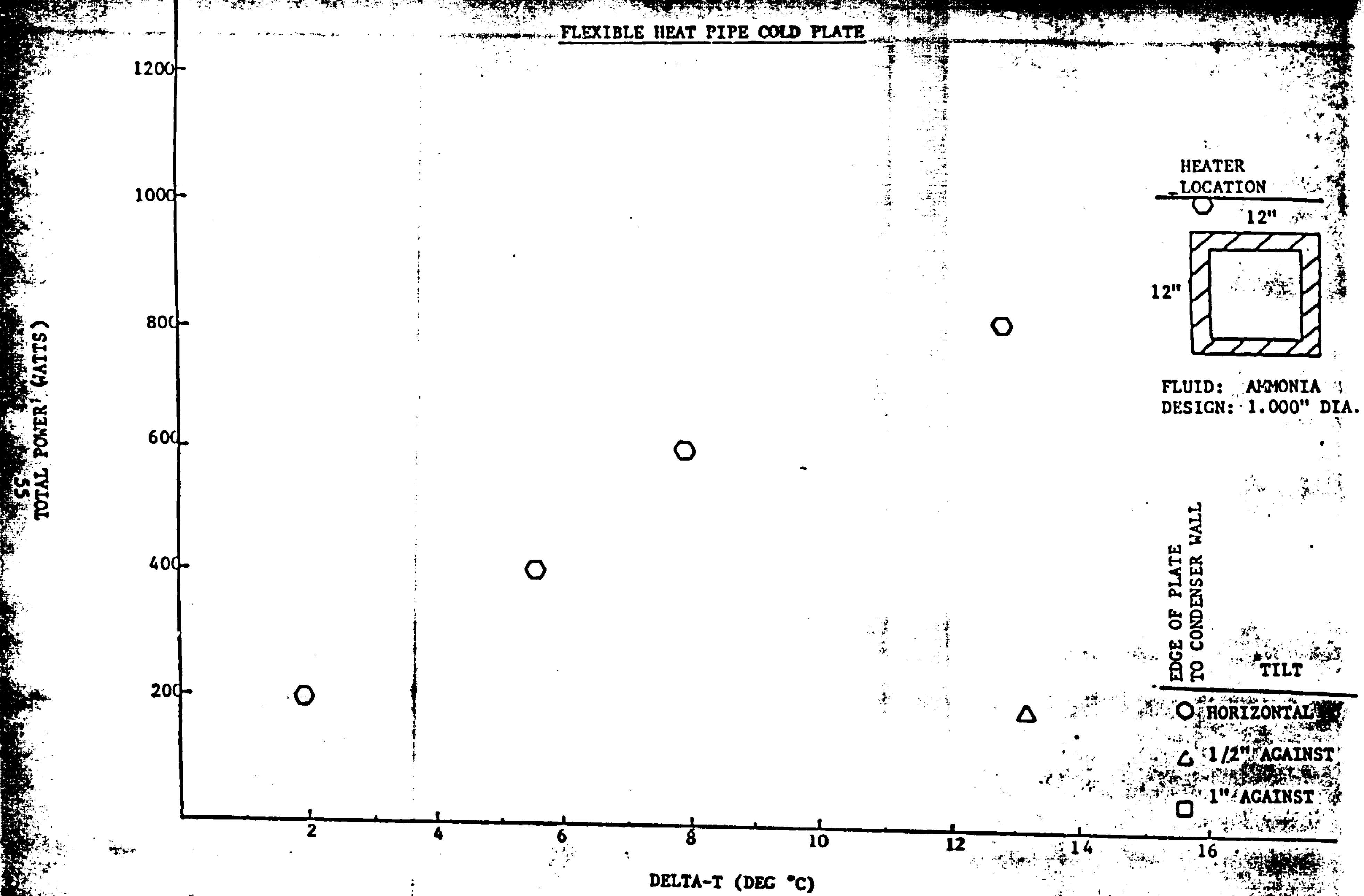


Figure 30. Performance vs. delta-T for several tilt angles

values for the FHPCP are calculated from the contact resistance determined for a tapered geometry. Actual testing of the FHPCP with the tapered condenser and thermal bus receptacle were conducted in Task 6. A complete description of the tapered interface design and test results are presented in Section 6.0 of this report.

Figures 31-34 show plots of the performance tests for ammonia. Tests were conducted at 0.00 cm, 1.27 cm and 2.54 cm inclinations against gravity using the four heater geometries on the cold plate surface. Testing at larger inclinations with ammonia yielded unsuccessful results due to the arteries depriming at 5.08 cm and 10.16 cm tilts.

The depriming of the arteries may be caused by flux material introduced during the sintering process. These materials could affect the wetting angle which will lead to poor capillary performance. These materials also appear to generate non-condensible gas within the arteries and cause them to deprime. Once this happens, the evaporator portion of the cold plate dries out and the plate stops transferring power.

Under Air Force Contract No. F33615-84-C-3415, similar performance was indicated when the pipe was initially charged with ammonia. However, by operating the pipe gravity aided for greater than 1000 hours and periodically venting, the flux materials appear to be removed by chemical reactions with the circulating ammonia.

An alternative flux material is being evaluated by Thermacore on an IR&D program. This material can be used to sinter aluminum at lower temperatures. Life tests to date show no incompatibility with ammonia. Also, the wick permeability is approximately a factor of 10 or better for the same pore size. Using this material, the cold plates will have similar delta-T limitations in the present wick structure, but will be able to transfer more power.

Taking into account the delta-T between the condenser of the FHPCP and thermal bus receptacle using the contact resistance discussed previously, a heat transfer capability of 750 W is achievable and falls within the 10°C delta-T limitation. Three of the four heater configurations were able to achieve this heat transport capability. These heaters provide a uniform heat load over the evaporator area. Strip heaters work best when they are positioned normal to the direction of the heat pipes.

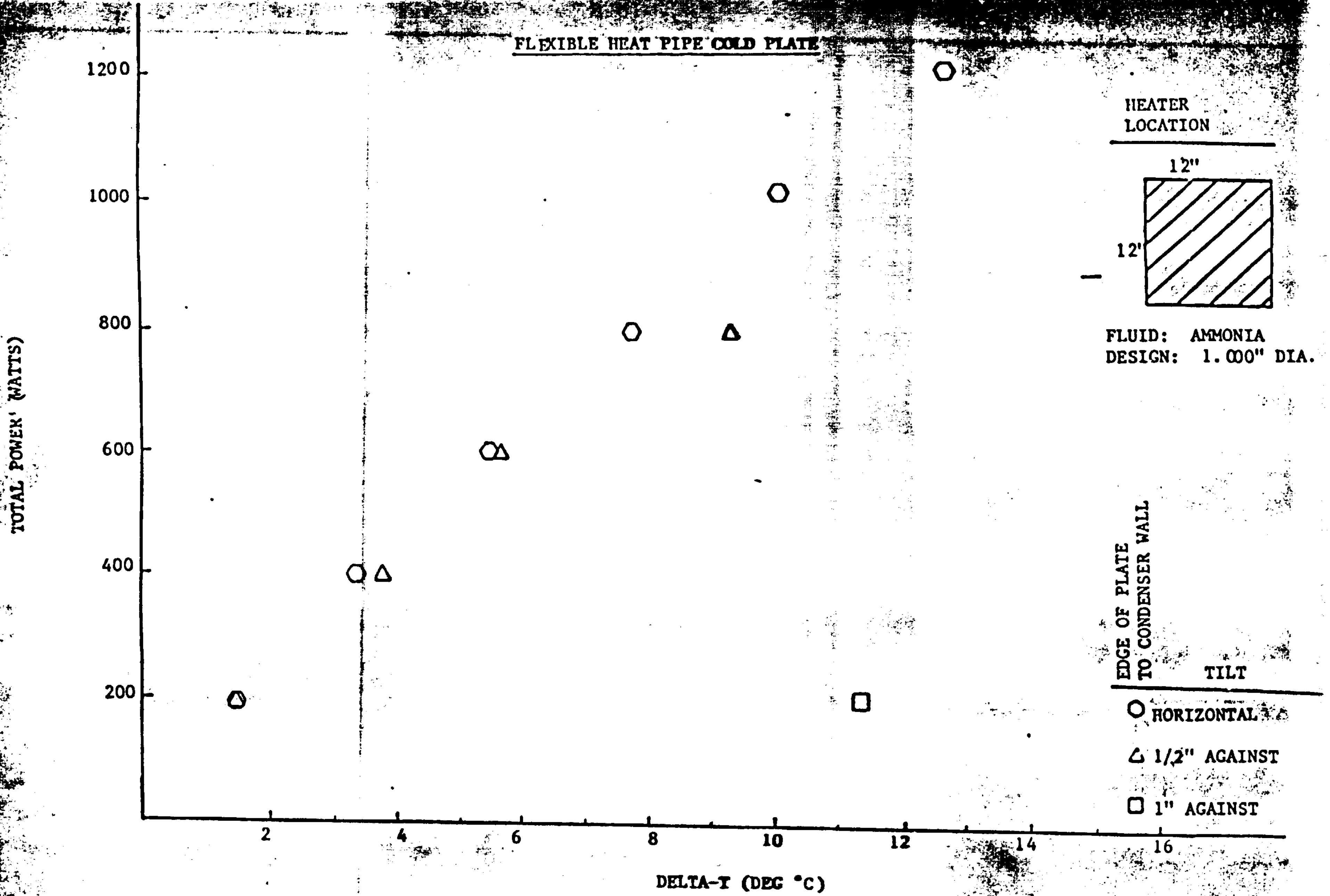


Figure 31. Performance vs. delta-T for several tilt angles

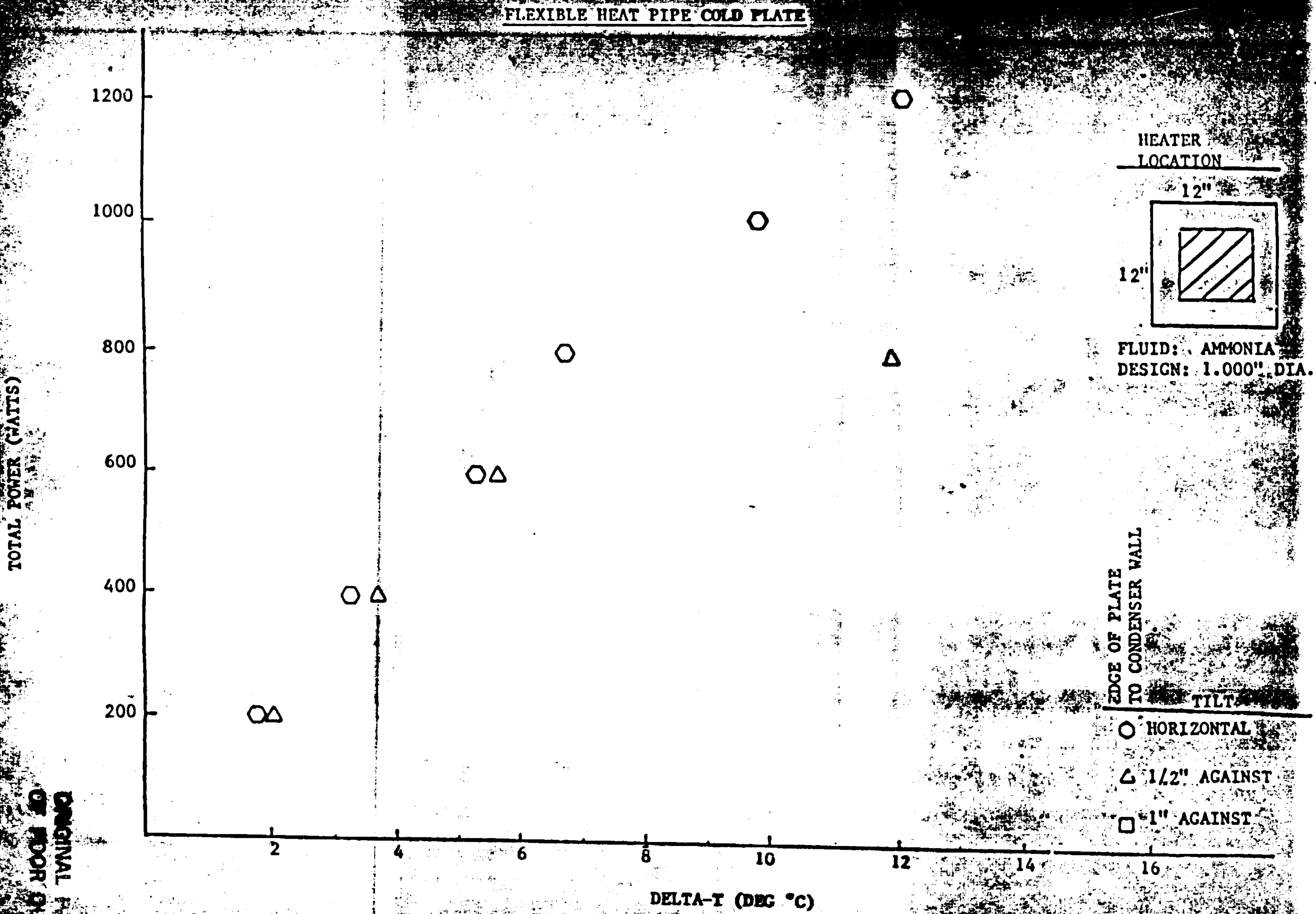


Figure 32. Performance vs. delta-T for several tilt angles

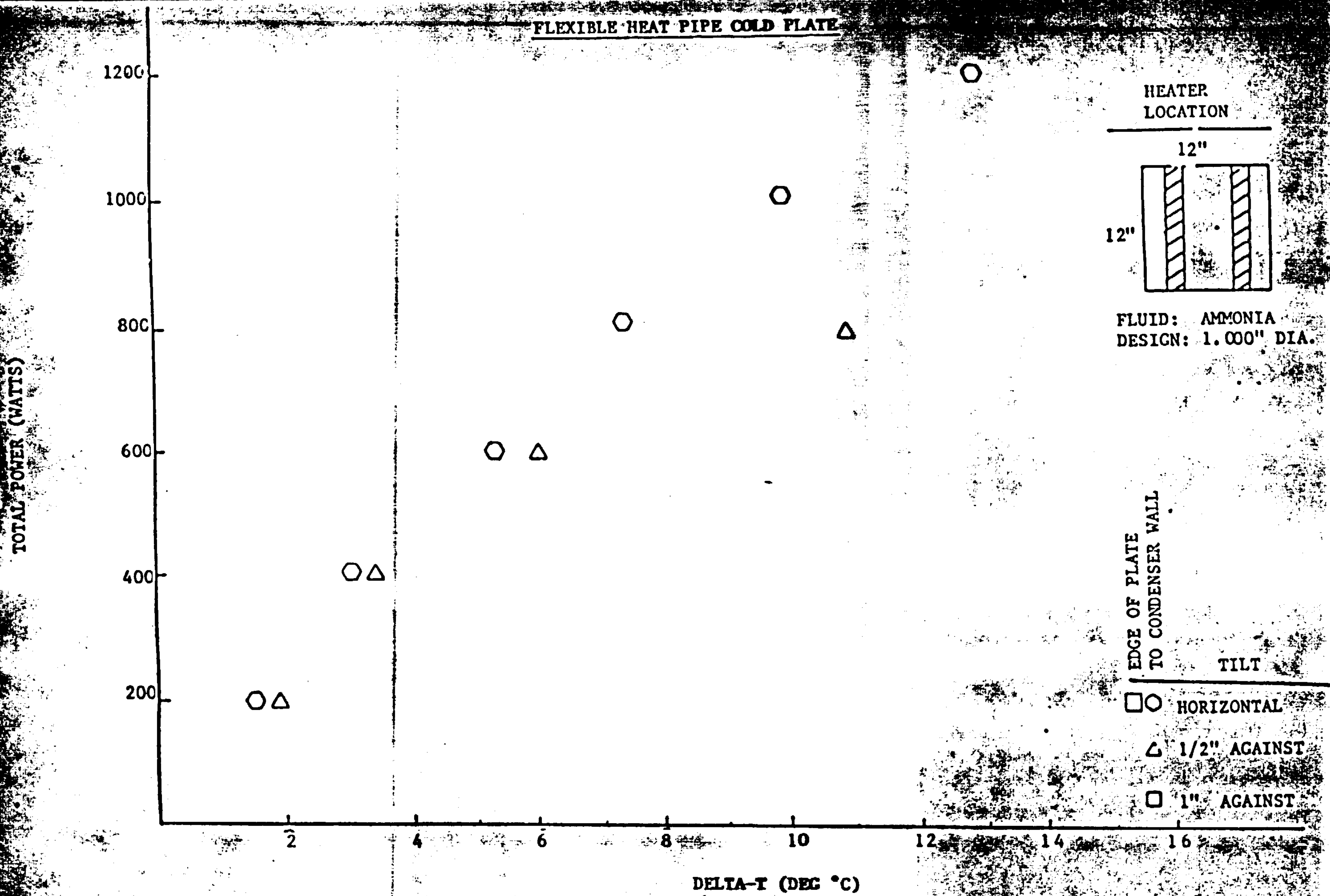


Figure 33. Performance vs. delta-T for several tilt angles

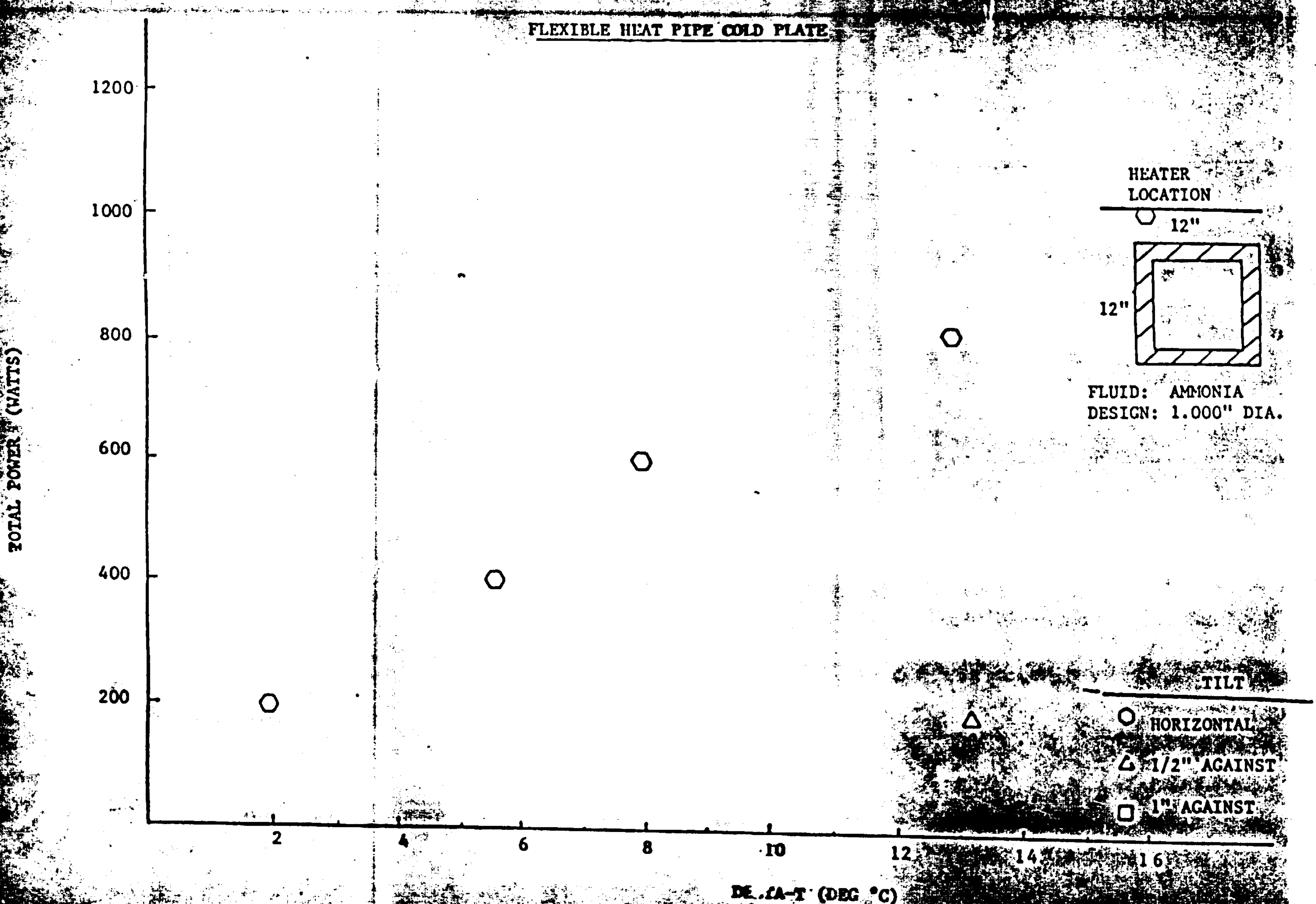


Figure 34. Performance vs. delta-T for several tilt angles

Performance testing of the FHPCP with 1.59 cm diameter holes demonstrated limited performance. Testing with acetone as the working fluid at a 0.00 cm inclination yielded a maximum power capability of 300 W. Testing at further inclinations against gravity proved unsuccessful. Due to the limited performance with acetone, ammonia was not tested.

Analysis of the data and FHPCP design indicates that a capillary limit could cause the low power transport capability. A capillary limit occurs when the sum of the liquid, vapor, and gravitational pressure drops exceed the capillary pumping capability of the wick. The location of the unexpectedly high pressure drop could be in the artery manifold which supplies the 16 evaporator pipes with liquid drawn from the condenser. A more detailed analysis on the mechanics of supplying liquid to the 16 evaporator tubes evenly through a manifold needs to be conducted. Unfortunately, this falls outside the scope of work for this work effort.

5.6.1 Conclusions: Flexible Heat Pipe Cold Plate

- The FHPCP with 2.54 cm diameter holes performed as expected with acetone, however, it appears that it is not maintaining primed arteries with ammonia.
- The FHPCP with 2.54 cm diameter holes, using acetone as the working fluid, performed best with the heat loading uniform over the evaporator area. Strip loads normal to the heat pipes also proved successful.
- The FHPCP with 2.54 cm diameter holes, using ammonia as the working fluid performed best with all heater geometries except for the heater positioned around the perimeter of the evaporator.
- The FHPCP with 1.59 cm diameter holes did not operate as designed.

5.6.2 Recommendations: Flexible Heat Pipe Cold Plate

- The FHPCP with 2.54 cm diameter holes will be the selected design to be delivered to NASA.

6.0 THERMAL BUS ALTERNATIVES

High performance thermal bus receptacles (TBR) are required to transfer the heat collected by the clamp-on cold plates and FHPCPs to the Space Station thermal bus heat transfer system. The receptacles can be placed in the thermal bus line for modularity, expansion and contingency. Figure 35 shows a schematic of how the thermal bus receptacles can be utilized in the thermal bus. The TBR can accept the clamp-on cold plate on its outer cylinder, and accept a FHPCP by plugging it into the inner cylinder.

6.1 THERMAL BUS RECEPTACLE DESIGN REQUIREMENTS

Both of the TBRs were designed based on the heat pipe design requirements listed in Table 11 below.

TABLE 11. Thermal Bus Receptacle Design Requirements

Thermal Bus Receptacle Working Fluid	Ammonia
Thermal Bus Receptacle Operating Temperature	0 - 40°C
Mass	Minimize
Outside Diameter (to accept the clamp-on cold plate)	7.62 cm
Inside Diameter (to accept to FHPCP condenser)	TBD
Length	TBD
Heat Transfer Capacity (outside diameter)	>1000 W
Heat Transfer Capacity (inside diameter)	>1000 W
Condensate Return Line Inside Diameter	$0.58 \text{ cm} \leq D_{CR} \leq 1.57 \text{ cm}$
Liquid Supply Line Inside Diameter	$0.45 \text{ cm} \leq D_{LS} \leq 1.27 \text{ cm}$

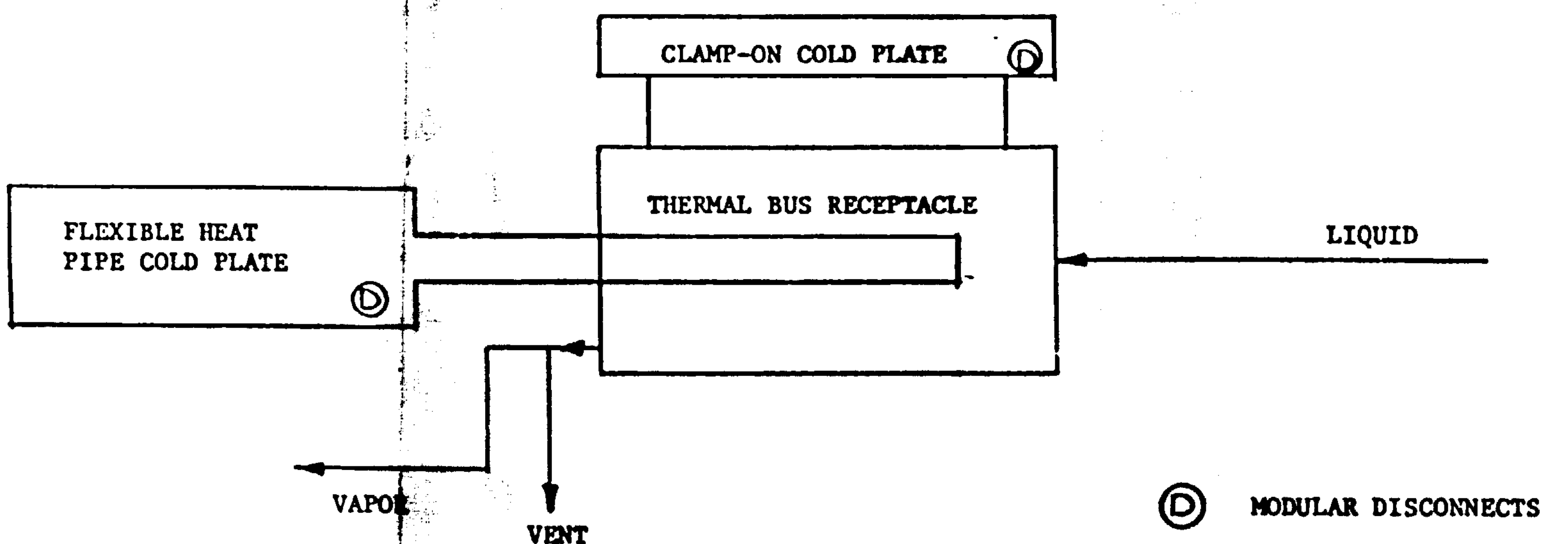


Figure 35. Integration of thermal bus receptacle flexible heat pipe cold plate and clamp-on cold plate to space station thermal bus

6.2 DESIGN

Three TBR concepts were established at the beginning of the Phase II program. Figures 36-38 show drawings of the three receptacles. Concept I is a flow through concept that can accommodate two phase flow exiting the receptacle. Concept II is a 100% liquid phase in - 100% vapor phase out concept which uses sensors and valves to regulate the amount of liquid delivered to the accumulator. Concept III is both a flow through and a phase separating receptacle concept.

Each of these concepts were evaluated in terms of the requirements listed in Table 11. In February, 1987, in a phone conversation with NASA personnel (E. Ferris, T. Brady) it was decided to pursue Concepts I and II only.

6.2.1 Load Sharing Interface Design

In Task 1 a delta-T requirement of 10°C maximum was established for a clamp-on cold plate and FHPCP. The delta-T was between the average evaporator temperature of these components and the thermal bus receptacle evaporator, taking into account the interface delta-T of the mating surfaces. Task 2 identified Indium foil as the interface medium which would provide the lowest interface delta-T for the clamp-on cold plate. In the design of the TBR's, an evaluation was conducted to determine the plug-in geometry between the condenser of the FHPCP and inner cylinder of the TBR which would provide a combined FHPCP, thermal interface delta-T < 10°C.

Two interface geometries were fabricated and tested for delta-T as a function of heat flux. The interface geometries include a straight and a tapered plug-in assembly. The straight plug-in assembly utilizes an interface material that provides a low delta-T and meets NASA outgassing requirements. The interface materials tested include thermal grease and a boron nitride/vacuum pump oil mixture.

The tapered plug-in interface geometry relies on the contact resistance between the mating components. Both mating surfaces were machined and polished to a surface finish of 16 rms or better. The delta-T was determined as a function of heat flux and contact pressure.

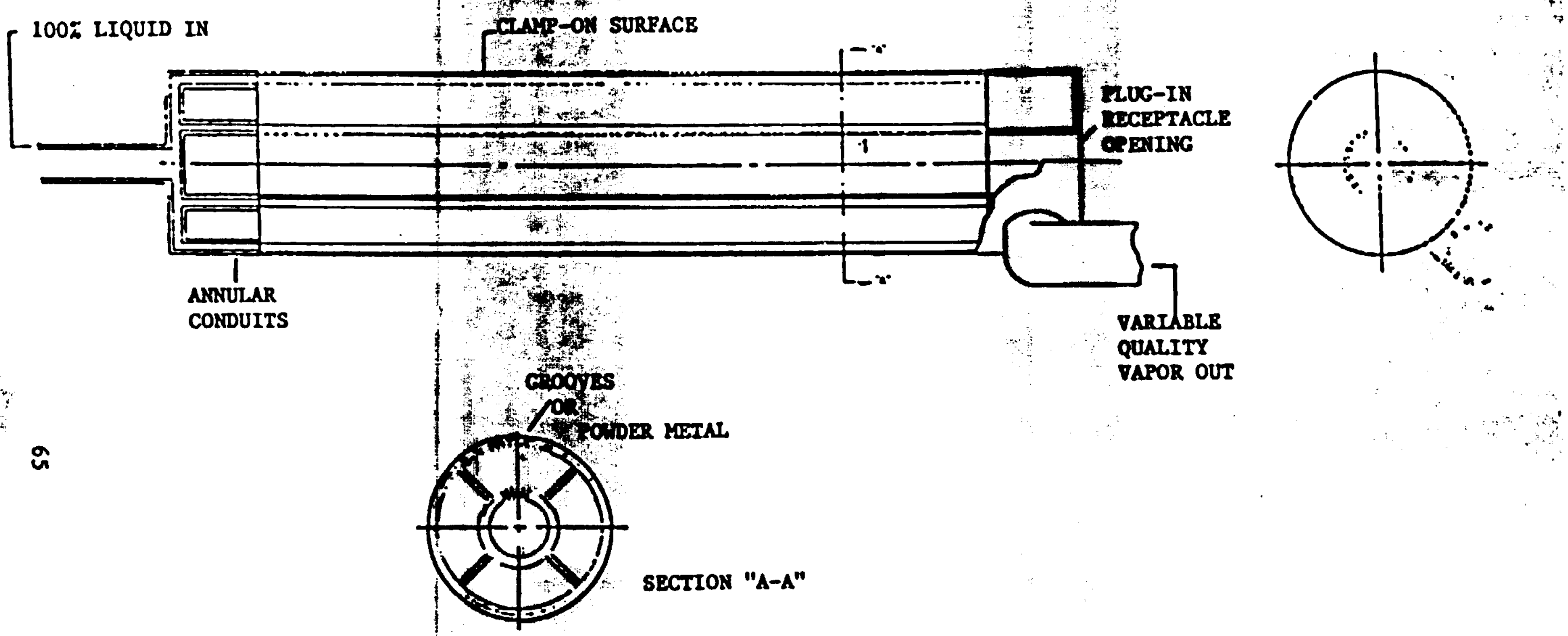


Figure 36. Thermal bus receptacle (concept I)

Concept I

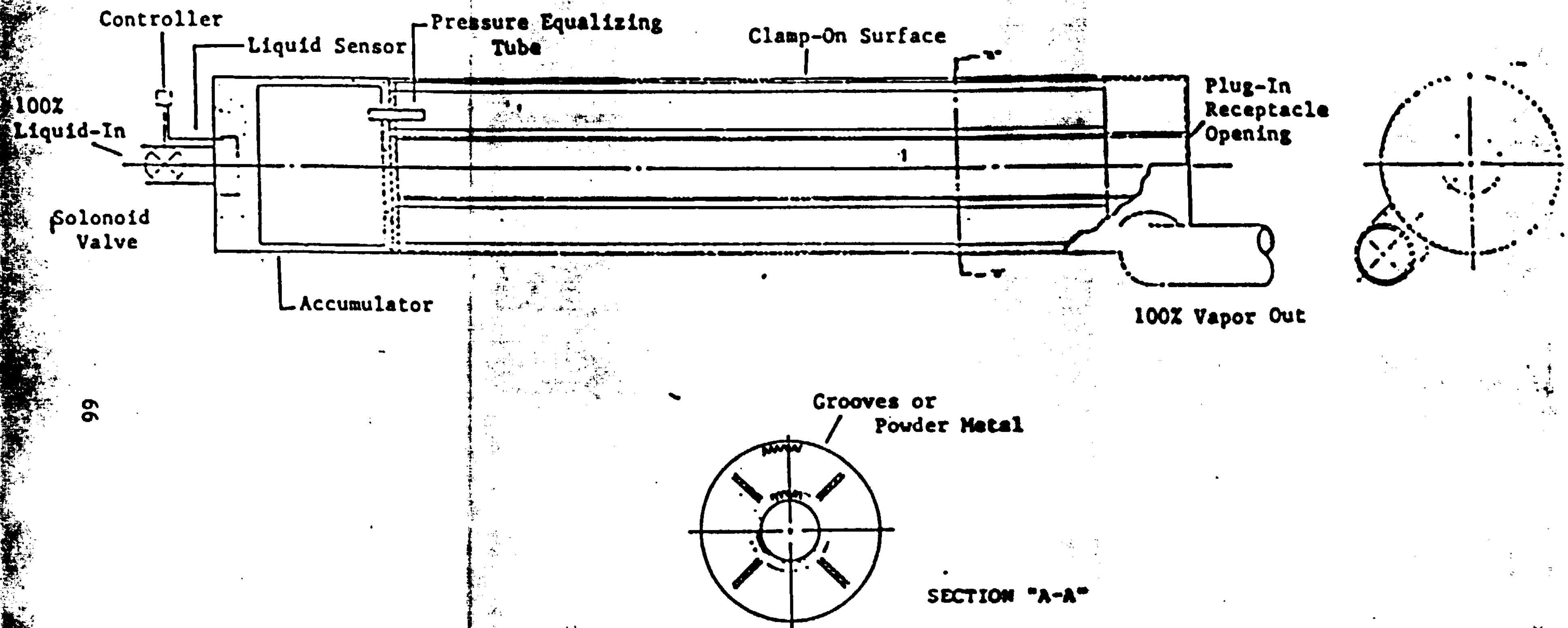


Figure 37. Thermal bus receptacle (concept II)

CONCEPT II

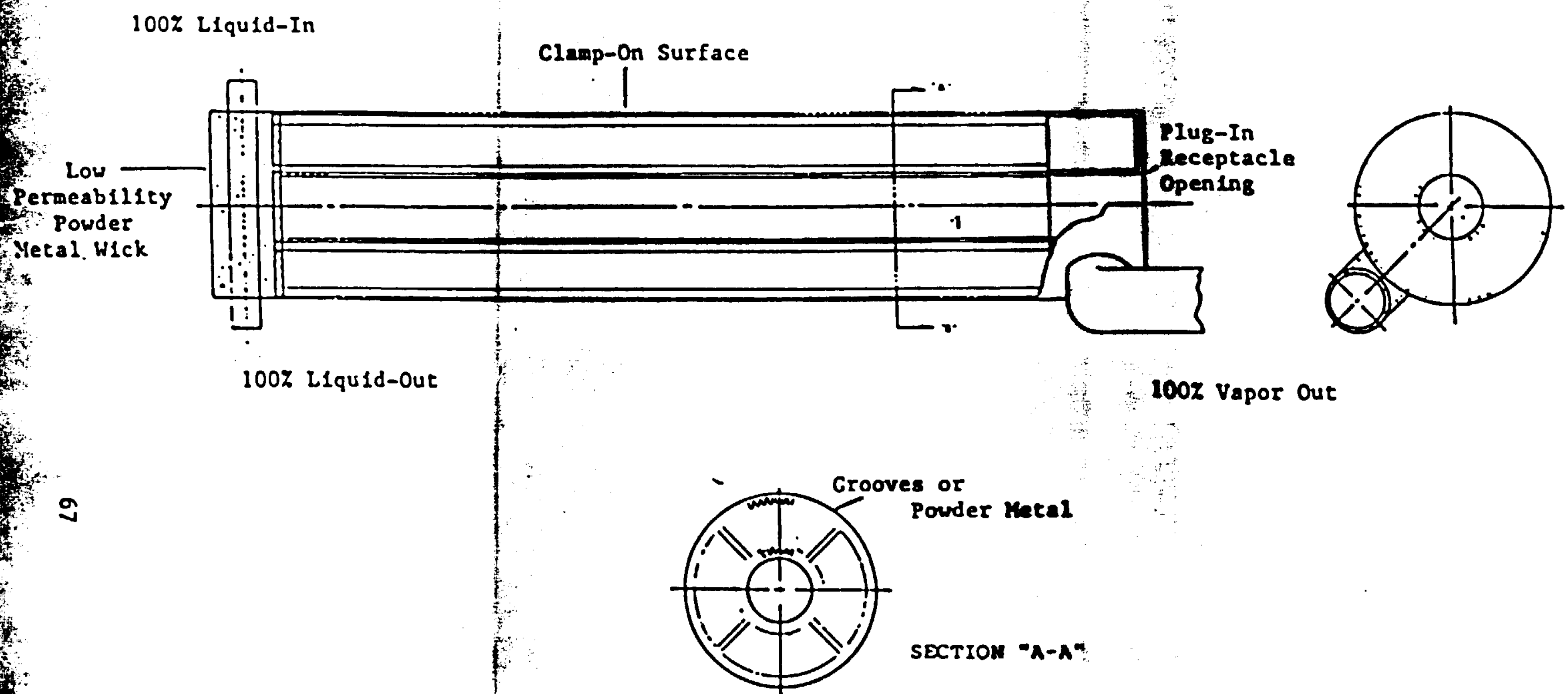


Figure 38. Thermal bus receptacle (concept III)

CONCEPT III

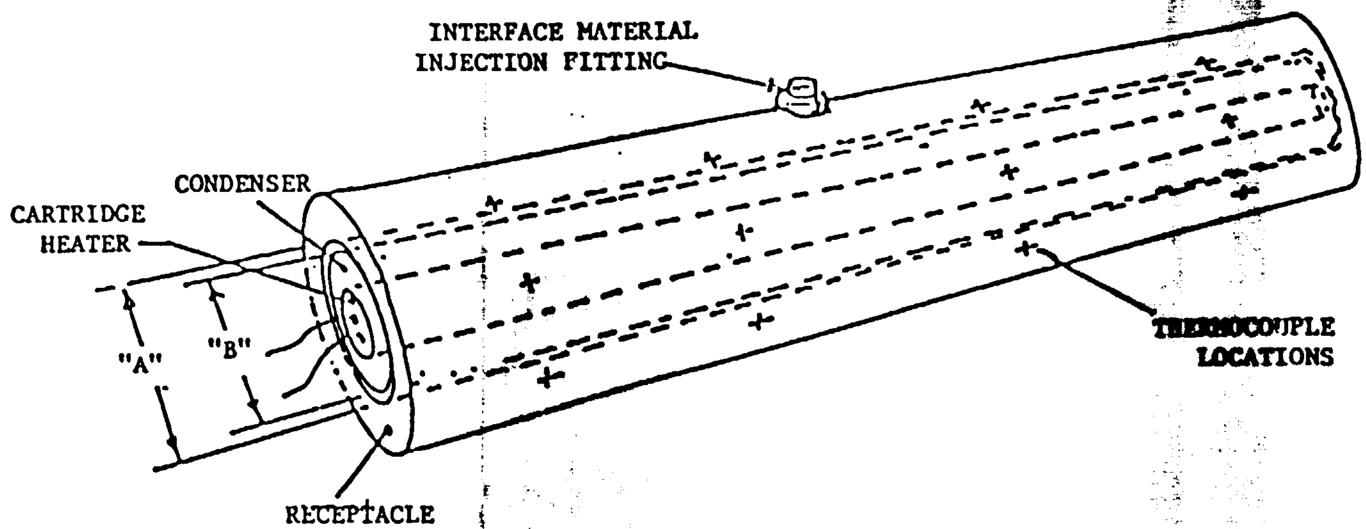
6.2.1.1 Description

A straight plug-in interface test article and a tapered plug-in interface test article were designed and fabricated. The straight plug-in interface test article utilized an interface material that was injected into the gap of three simulated flexible heat pipe cold plate condensers, each with a slightly different outside diameter. The condensers provided a 0.013 cm, 0.025 cm and a 0.051 cm gap between the mating components. A drawing of the straight plug-in interface test article is shown in Figure 39.

Two interface materials were tested; Emerson and Cuming TC4 thermal grease and a boron nitride/vacuum pump oil mixture. The Emerson and Cuming thermal grease material was selected due to its performance in Task 2 of this program in which a 1°C delta-T was achieved at 7 W/cm^2. The boron nitride/pump oil combination was selected due to its interface performance under NASA Contract NAS8-36263, "Heat Transport Across Structural Boundaries." Testing under this program yielded a 1.5°C delta-T at 3.5 W/cm^2 for the boron nitride/pump oil combination.

The tapered plug-in interface test article relies on the contact resistance between the flexible heat pipe cold plate condenser and the thermal bus receptacle to produce a low delta-T. A standard Jarno taper, 0.600 inch/ft was used for this application. A drawing of the tapered plug-in test article is show in Figure 40. Testing for interface delta-T and ease of disassembly was conducted at several contact pressures. To help disassembly, a thin film of diffusion pump oil was applied to the tapered components.

Heat was applied to the interface test articles (straight and tapered) by a 1.27 cm cartridge heater inserted in the center of the simulated condenser. The heat was transferred radially outward, where it was removed by cooling water. Thermocouples located in the wall of the simulated condenser and in the wall of the simulated thermal bus receptacle were used to determine the delta-T across the thermal interface.



CASE	DIM. "A" RECEPTACLE I.D. (cm)	DIM. "B" CONDENSER I.D. (cm)	GAP SIZE (cm)
1.	3.251	3.226	.003
2.	3.251	3.200	.025
3.	3.251	3.15	.051

Figure 39. Thermal bus receptacle - straight plug-in concept test apparatus

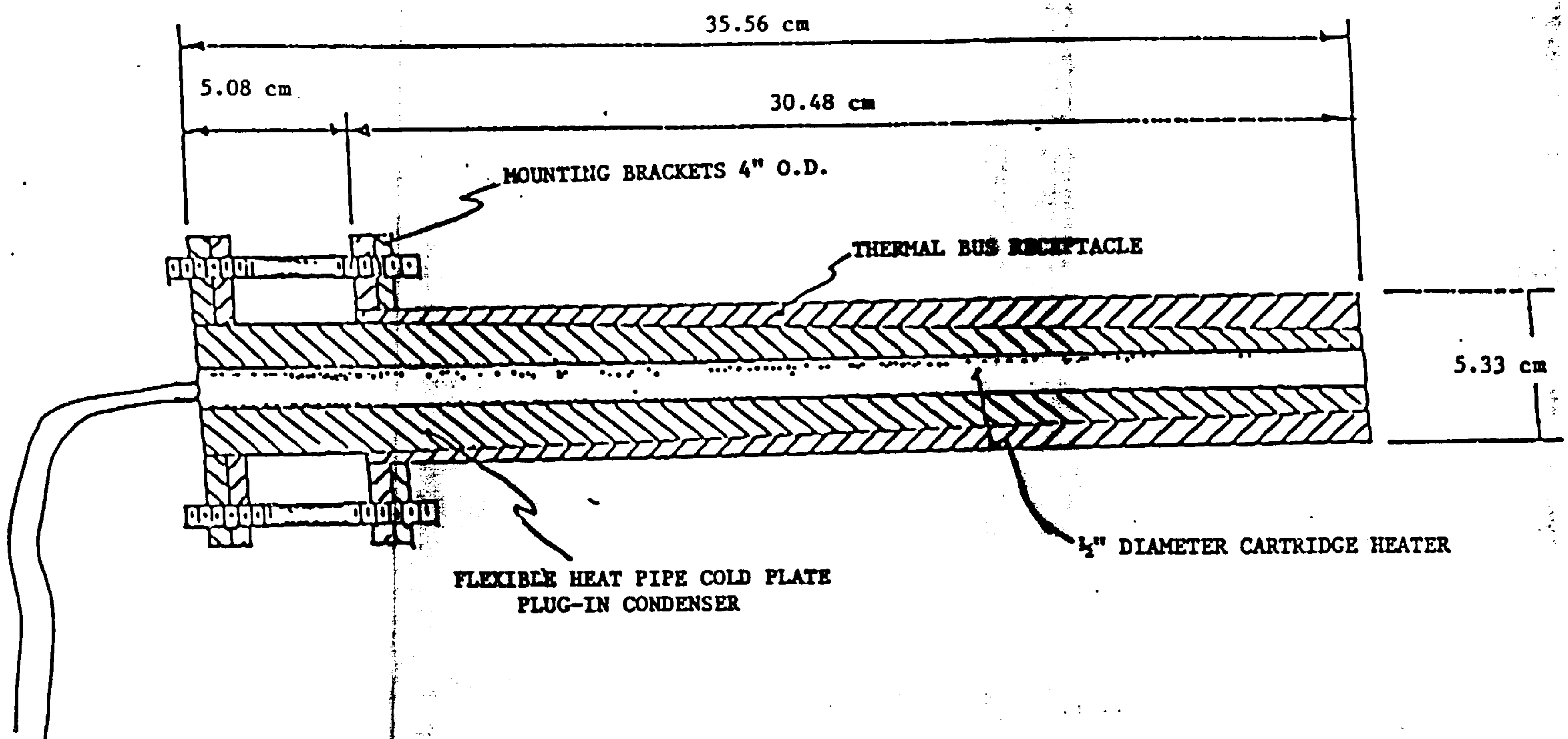


Figure 40. Thermal bus receptacle - tapered plug-in test apparatus

6.2.1.2 Thermal Interface Test Procedure

The following procedures were used to collect the thermal interface test data for the straight and tapered plug-in assemblies.

Straight Plug-In Test Article

A) Insert the simulated condenser into the simulated receptacle

B) Adjust the gap size to the desired distance using metal foil of known thickness.

C) Inject the interface material into the interface gap.

D) Insert thermocouples into the simulated condenser wall.

E) Place a coating of gasket sealant around the bottom gap circumference to prevent water leakage.

F) Place the test article into circulated cooling water. Turn on the power to 100 W.

G) Allow the test article to reach steady state and record the thermocouple readings into the laboratory notebook.

H) Increase the electrical power in 100 W increments and repeat steps G-F up to 1000 W.

Tapered Plug-In Test Article

A) Insert the simulated tapered condenser into the simulated receptacle.

B) Bolt the two components together with the retaining rings.

C) Torque the bolts to meet the required contact pressure.

D) Place the test article into circulating cooling water. Turn power on to 100 W.

E) Allow the test article to reach steady state and record the thermocouple readings into the laboratory notebook.

F) Increase the electrical power in 100 W increments and repeat steps E-F up to 1000 W.

6.2.1.3 Thermal Interface Test Data

The test data taken for the straight and tapered plug-in test articles are plotted in Figures 41-43. The data for the straight plug in test article are shown as a plot of delta-T versus thermal power transferred for thermal grease and the boron nitride/pump oil mixture. The data for the tapered plug-in test article are shown as a plot of delta-T versus contact pressure.

Test data for the boron nitride/pump oil mixture indicate delta-T's above predicted values. These results are attributed to the boron nitride and vacuum pump oil separating during injection into the interface gap producing voids between the individual boron nitride particles. This increases the contact resistance between the particles and in turn increases the delta-T.

Test data for the thermal grease also indicated a delta-T above predicted values. The results were attributed to the inability to inject the thermal grease into the narrow gap with sufficient distribution to completly fill the gap. As with the boron nitride/pump oil mixture, this would produce a high contact resistance between the walls of the mating components.

Data analysis also indicated a non-uniform temperature profile along the male plug-in component length. This is possibly related to surface irregularities between the cartridge heater and the hole in which it was inserted.

Test data for the tapered interface geometry indicated a 3.7°C delta-T at 1000 W which corresponds to a contact resistance of 1.35 cm^2-°C/W. This took place at a contact pressure of 200 lb/in^2. According to test data already presented for the FHPCP with acetone as the working fluid, 800 W is achievable with a 10°C delta-T which includes the interface delta-T.

Following the interface delta-T testing, the tapered assembly was placed in a vacuum chamber at 1×10^{-5} torr to simulate a space environment. The mating components were torqued to produce an interface pressure of 300 lb/in^2 at 40°C. This test was conducted to determine if diffusion bonding will occur between the highly polished components. Prior to assembling the tapered joint, a thin film of diffusion pump oil was applied to the mating surfaces to act as a mold release. The test was terminated after accumulating 1002 hours.

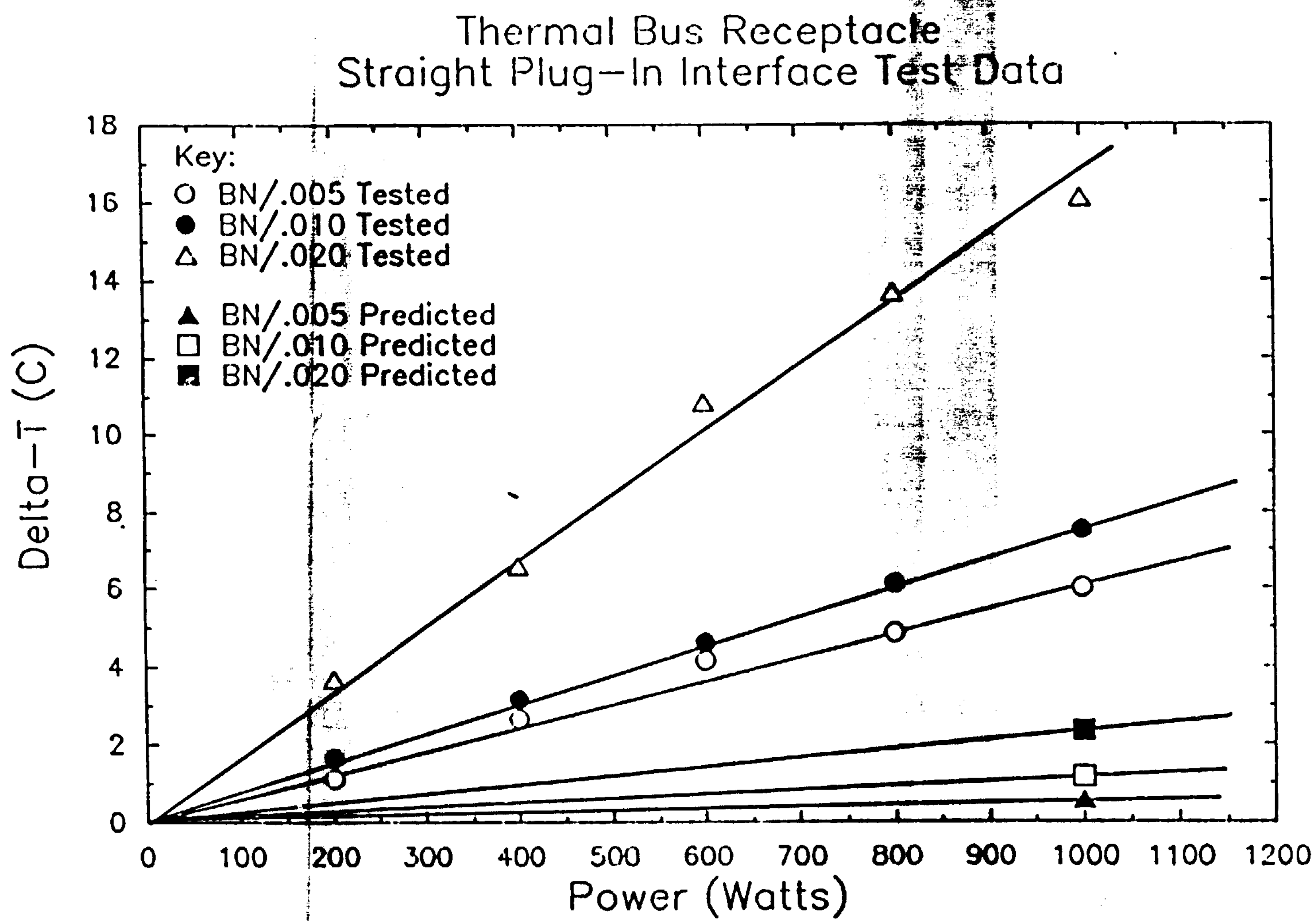


Figure 41. Thermal bus receptacle - straight plug-in interface test data

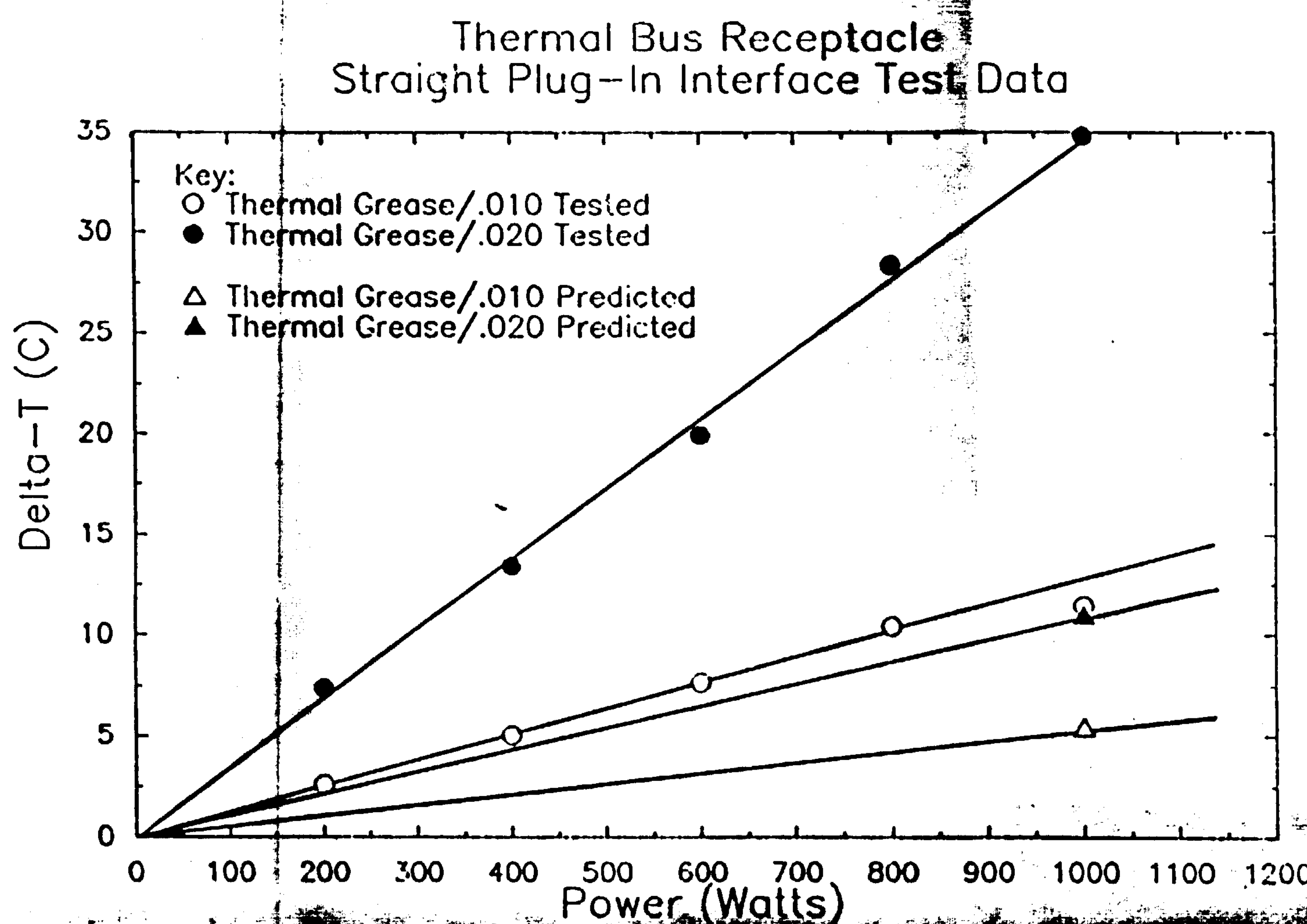


Figure 42. Thermal bus receptacle - straight plug-in interface test data

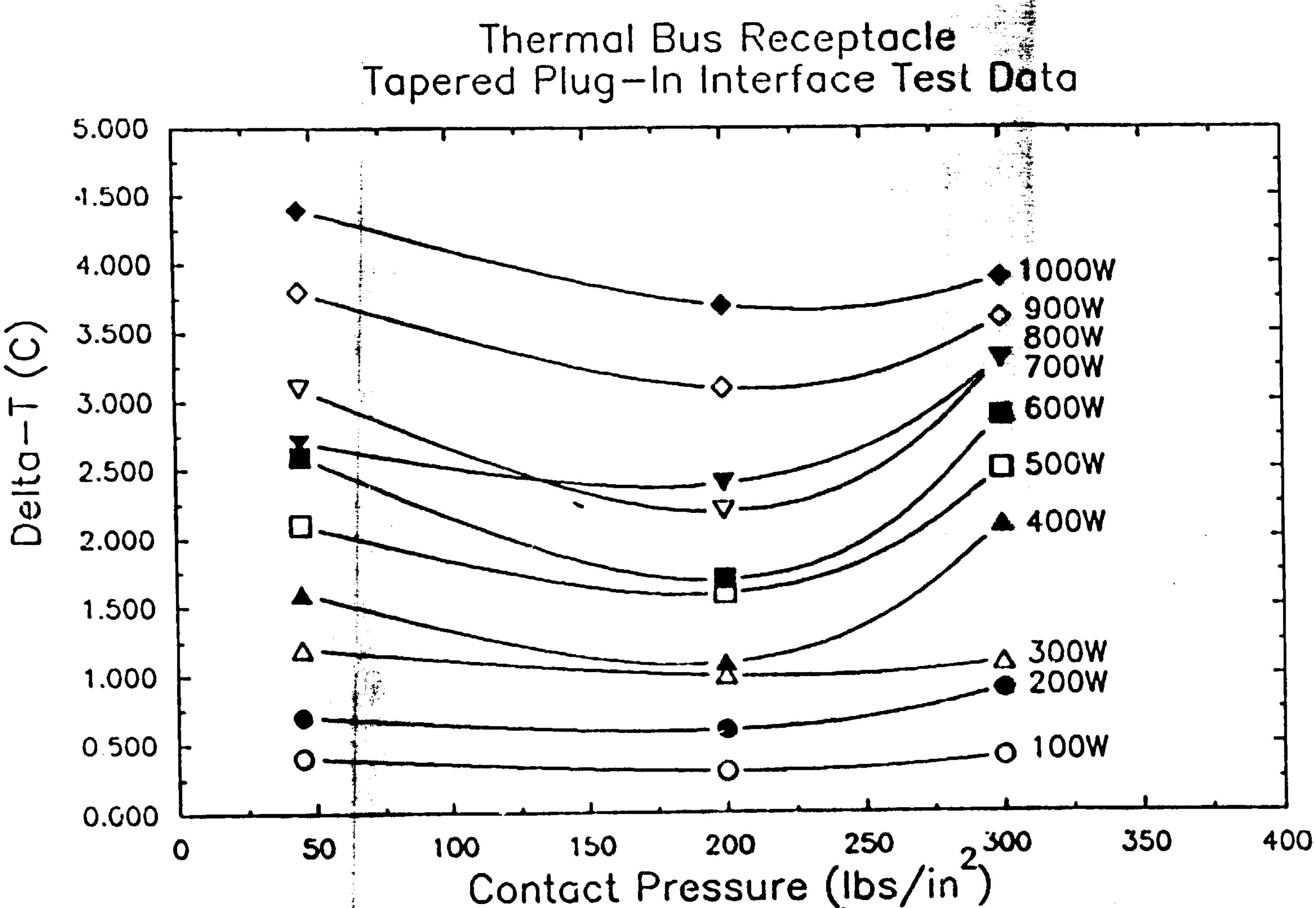


Figure 43. Thermal bus receptacle - tapered plug-in interface test data

Dismantling of the tapered joint was accomplished with minor resistance. Stainless steel nuts were use to jack the simulated condenser out of the simulated receptacle. This was accomplished by turning one nut at a time approximately a quarter of a turn. The pattern in which the retracting nuts were turned was the standard pattern for mounting flanges in which gaskets are used for seals.

Five passes around the flange were required to remove the simulated condenser from the simulated receptacle. Inspection of the interface components indicated that diffusion pump oil was still present in the interface gap.

The acceptable results found during delta-T testing and diffusion bond testing has made the tapered interface geometry the preferred design for joining the FHPCP to the TBR. This design also reduces the time required and risk associated with working outside of the habitat modules, since a thermal compound need not be applied to the joint.

6.2.2 Thermal Bus Receptacle - Wick Structure Design

The wick structure design is separated into two sections. The first section outlines the wick structure design for TBR Concept I, variable quality vapor out. The second section outlines the wick structure for TBR Concept II, 100% liquid in phase - 100% vapor out phase.

6.2.2.1 Wick Structure Design - Concept I

The wick structure for the variable quality vapor out TBR focused on mass, heat transport capacity and fabricability. An axial grooved wick structure was selected as the preferred design. Axial grooves, 0.15 cm deep x 0.07 cm wide with a 0.07 cm land, have sufficient capillary pumping capability to transport the required loads.

The axial grooves for the outer cylinder of the TBR were extruded from 1100 aluminum. A drawing of the outer cylinder and a photograph of the extrusion are found in Figure 44. The axial grooves for the inner cylinder were machined from 1100 aluminum. A drawing of the inner cylinder and a photograph of the machined part are shown in Figure 45. The inner cylinder was also designed to accept the tapered condenser from the FHPCP, Figure 46.

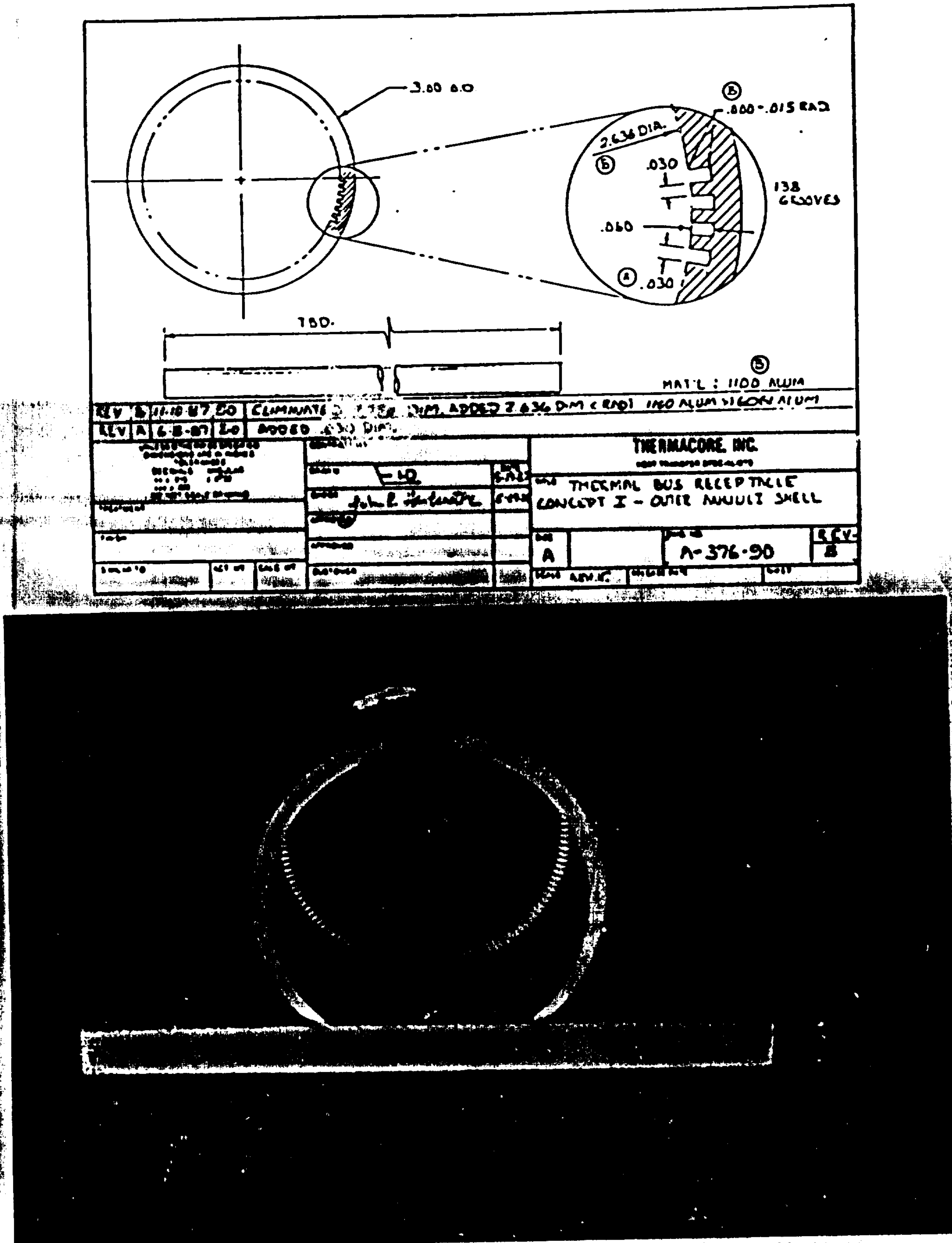
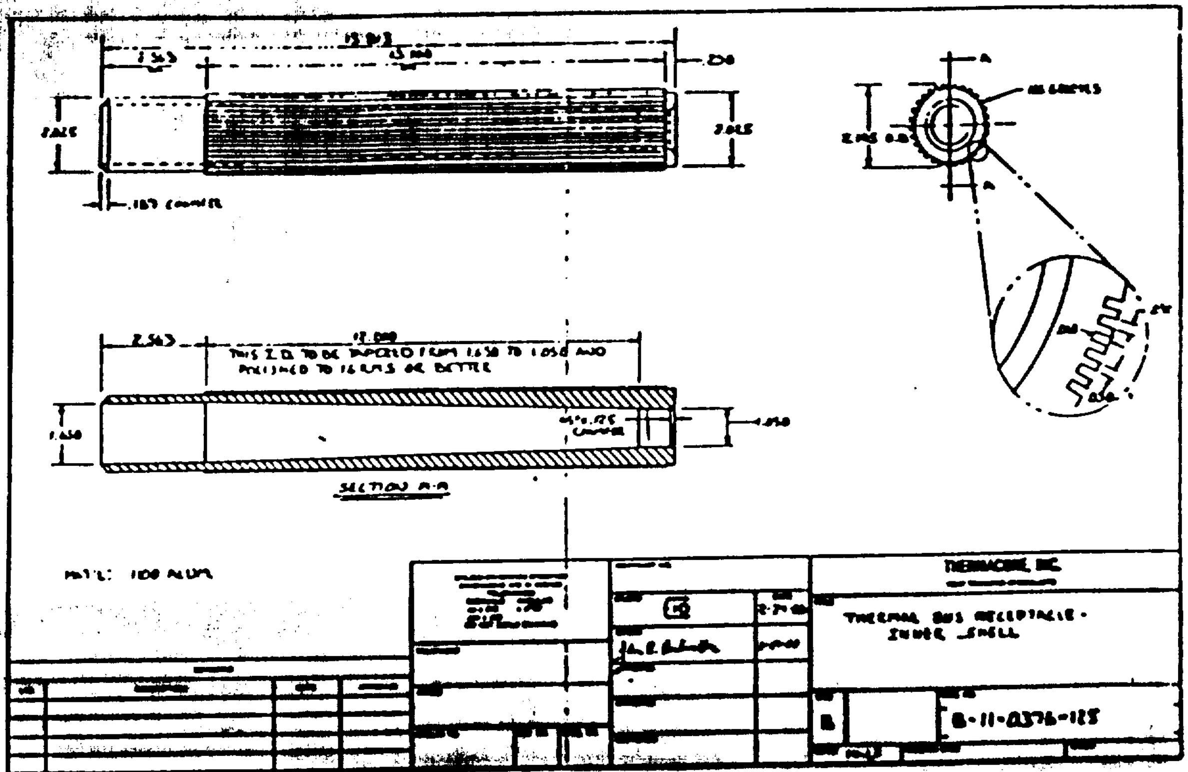


Figure 44. Thermal bus receptacle concept I outer diameter wick structure



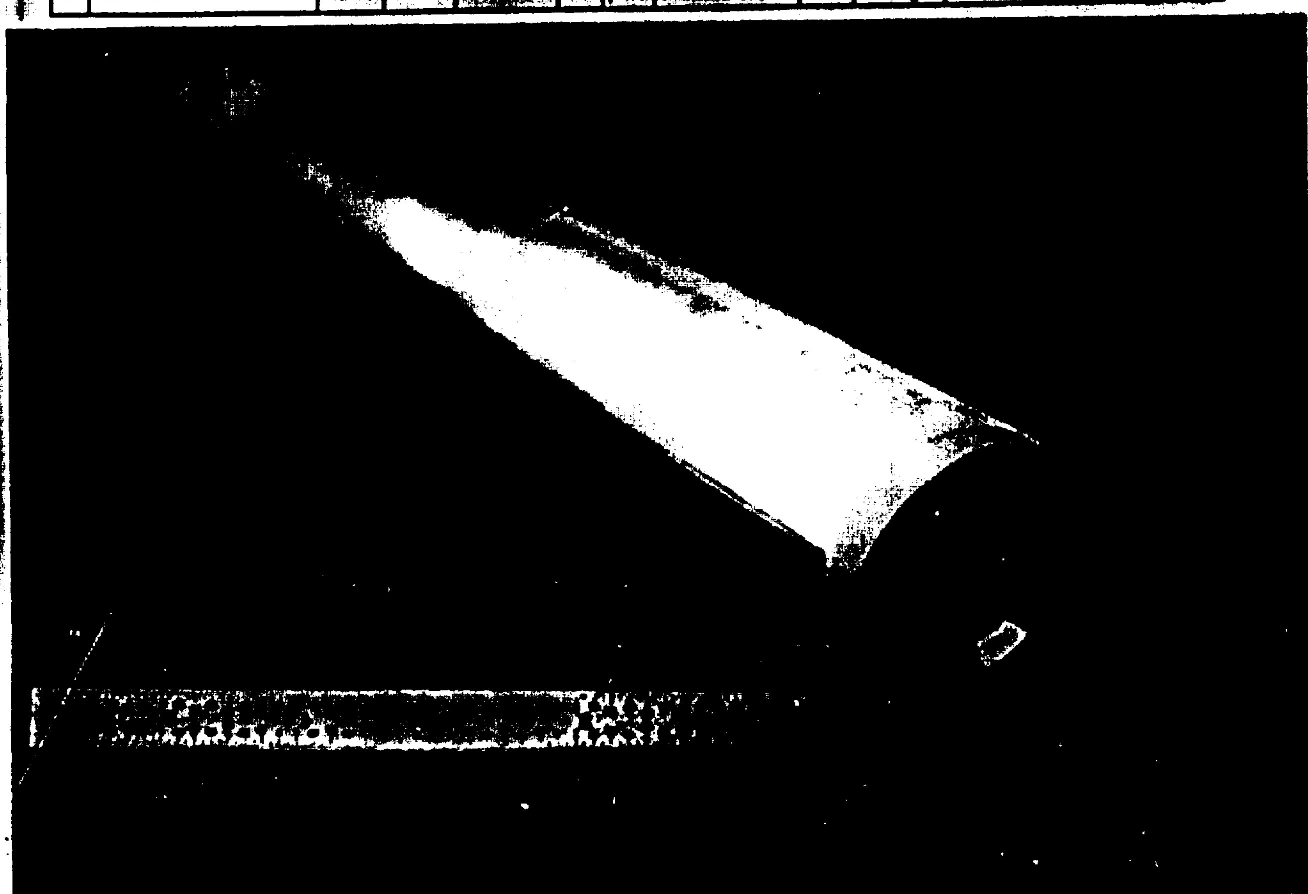


Figure 45. Thermal bus receptacle concept I inner diameter wick structure

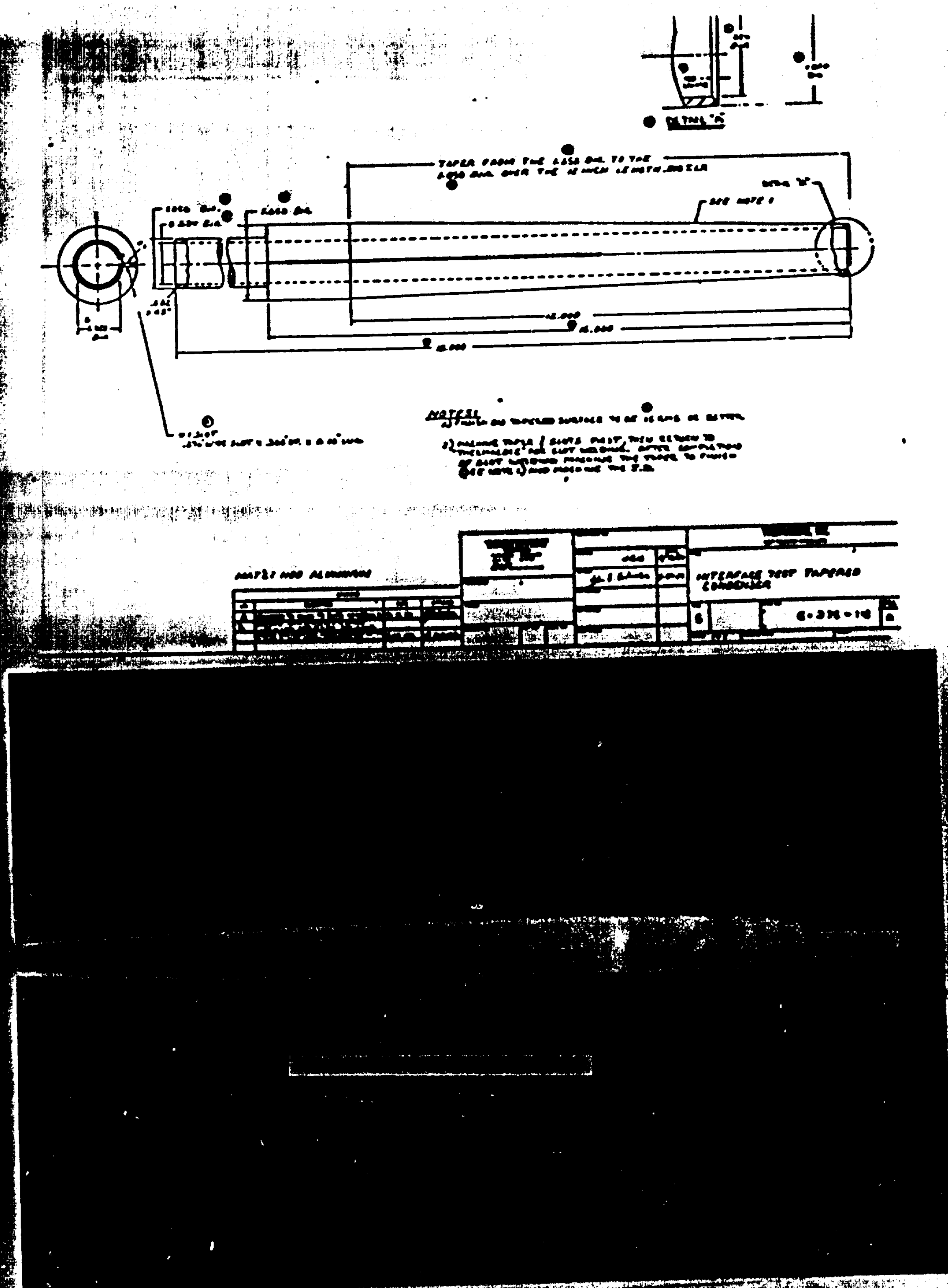


Figure 46. Flexible heat pipe cold plate tapered condenser

In order to direct the supply of liquid to the grooves from the liquid supply line, a liquid distribution plate placed in the TBR, was designed to allow for liquid to supply the grooves only. An aluminum powder metal liner was sintered in front of the liquid portion plate and directly against the axial grooves in order to establish a resistance to the ammonia flow as to assure all grooves are supplied with fluid at the same time. This resistance to the flow is needed to overcome the affects of the earth's gravity on the ammonia working fluid. For a space application, the powder metal wick would not be required. A drawing and a photograph of the liquid distribution plate in the TBR is shown in Figure 47.

In the actual application, ammonia for the Space Station thermal bus will be injected into the TBR through the liquid supply line. The ammonia will feed the axial grooves and will evaporate due to the waste heat generated from equipment and experiments attached to the clamp-on cold plate and FHPCP. The variable quality vapor will exit the TBR via the condensate return line and reject the waste heat by radiating to space from the Space Station radiators.

6.2.2.2 Wick Structure Design - Concept I

The wick structure for the 100% liquid phase in - 100% vapor phase out TBR utilizes sintered powder metal and liquid return arteries. The sintered powder metal will only wick in enough fluid to fill the wick structure. The wick structure is the same basic design used for the clamp-on cold plates and FHPCP. A coarse powder circumferential wick was sintered to the inside diameter of the outer cylinder and the outside diameter of the inner cylinder. Due to the geometry of the plug-in component of the TBR, a limited vapor space prevented the use of individual arteries for each wick structure. Arteries were sintered integral to the coarse circumferential wick on both cylinders. A drawing of the TBR Concept II wick structure is shown in Figure 48. Pressure drop calculations indicate that the wick structure has sufficient capillary pumping capability to accommodate the load sharing from both the clamp-on cold plate and FHPCP.

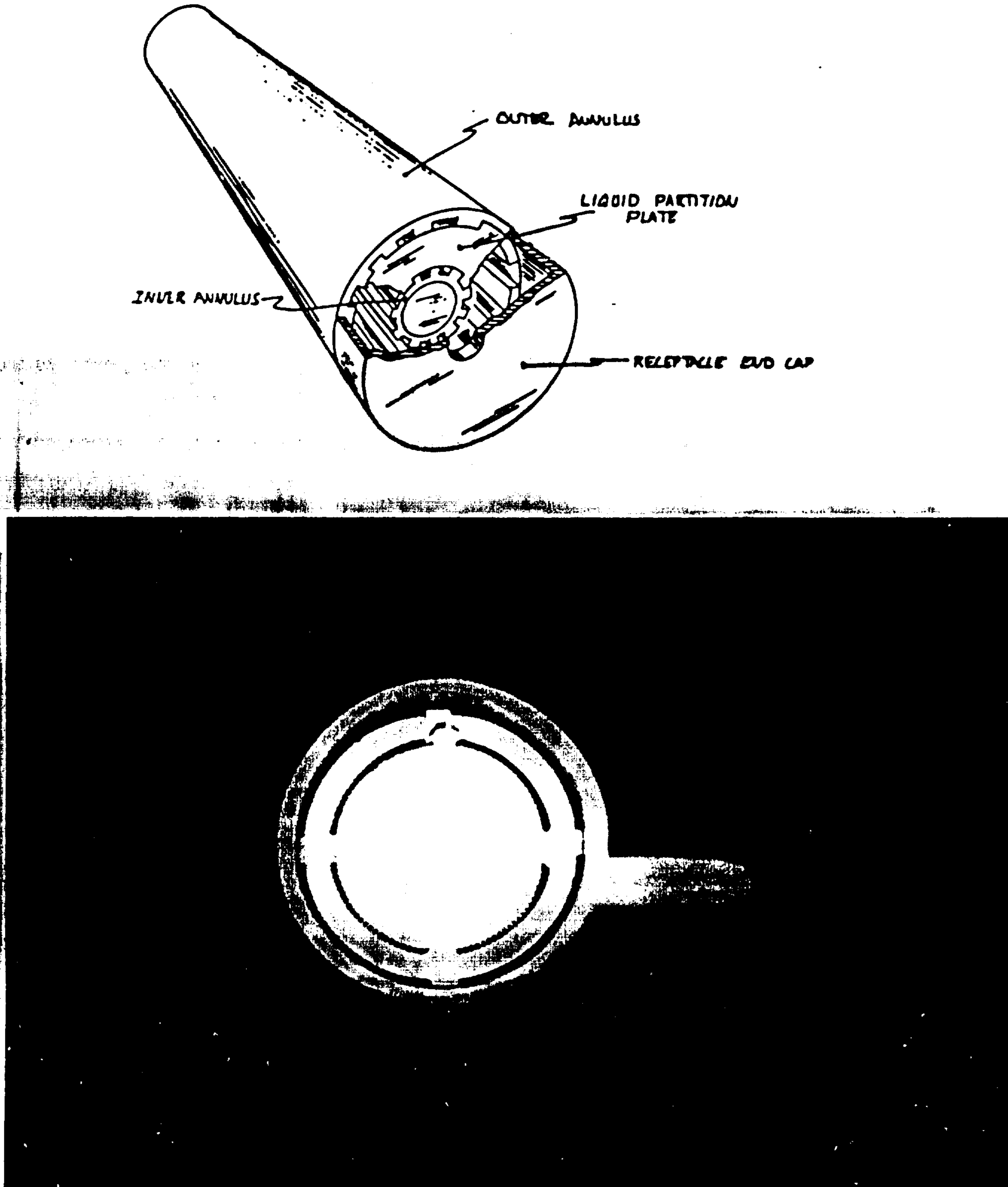


Figure 47. Thermal bus receptacle concept I wick structure with liquid partition plate

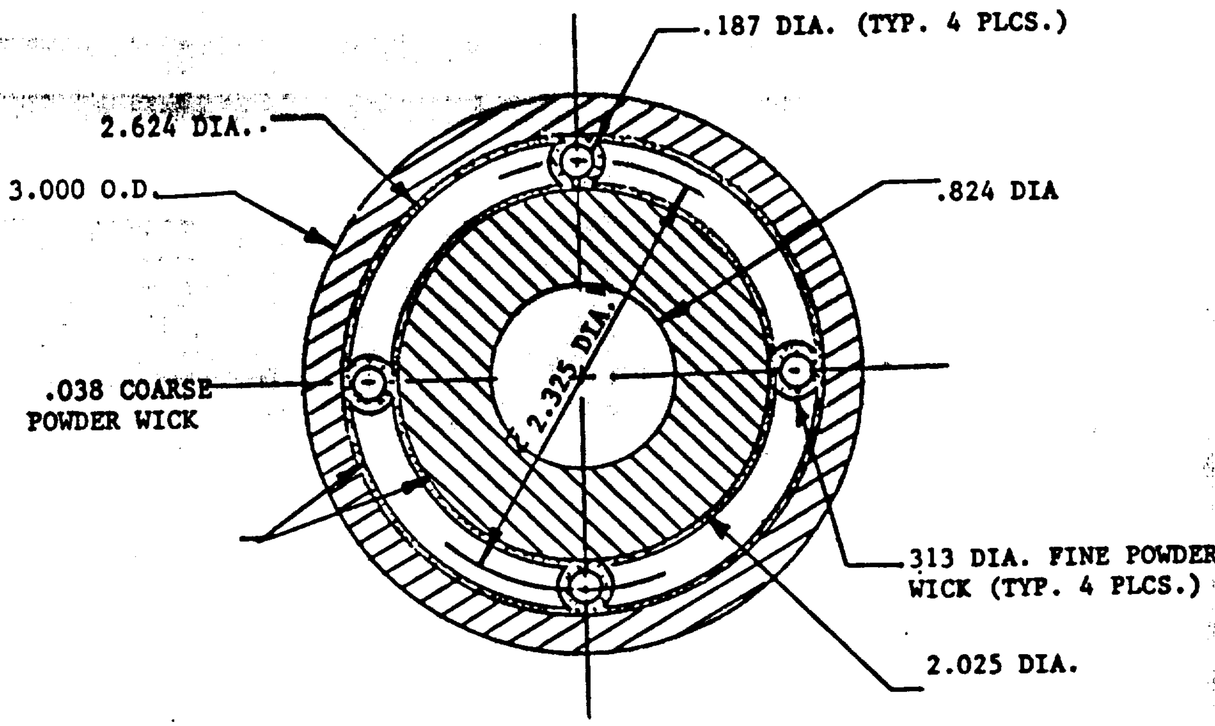


Figure 48. Thermal bus receptacle concept II wick structure

The 100% liquid phase in - 100% vapor phase out design utilizes a liquid reservoir or accumulator to supply liquid to the arteries and powder metal wick structure. A liquid level sensor is needed to determine the amount of ammonia in the reservoir so that the ammonia flow from the thermal bus can be regulated into the TBR during operation. Two liquid level sensors were evaluated for this application, a manual sensor and an automatic sensor.

Both the manual and automatic options were reviewed and it was determined that the automatic sensor would not be feasible. Two liquid level sensor manufacturers were contacted to determine if an automatic external ultrasonic sensor could detect liquid ammonia accurately. Both manufacturers indicated that due to the large diameter of the aluminum accumulator, the wall-wick interface and the porosity of the wick structure, an ultrasonic sensor would not operate properly.

The thermal bus receptacle is expected to interface with the main contractor's automatic level control system currently planned for the Space Station thermal bus. Thermacore therefore, utilized a manual sensor concept in the form of a sight glass and valve for the prototype thermal bus receptacle accumulator. This provided level control sufficient to demonstrate performance.

A powder metal wick structure is needed in the liquid reservoir to supply the wick of the heat transfer area with fluid. Fine powder metal (-325 mesh) was sintered into the reservoir such that the arteries of the heat transfer area would be supplied directly. Liquid from the reservoir is wicked into the powder metal and drawn into the arteries of the heat transfer area.

During operation, a pressure differential will exist between the vapor space of the heat transfer area and the accumulator. The pressure differential is due to the vapor pressure of ammonia being higher in the heat transfer area than the accumulator. This pressure differential will restrict liquid ammonia flow into the wick structure. To compensate for the pressure differential, a pressure equalizing tube was incorporated into the design. The tube was positioned between the vapor cavity of the heat transfer area and the vapor cavity of the accumulator. Figure 49 illustrates the accumulator with the pressure equalizing tube and wick structure.

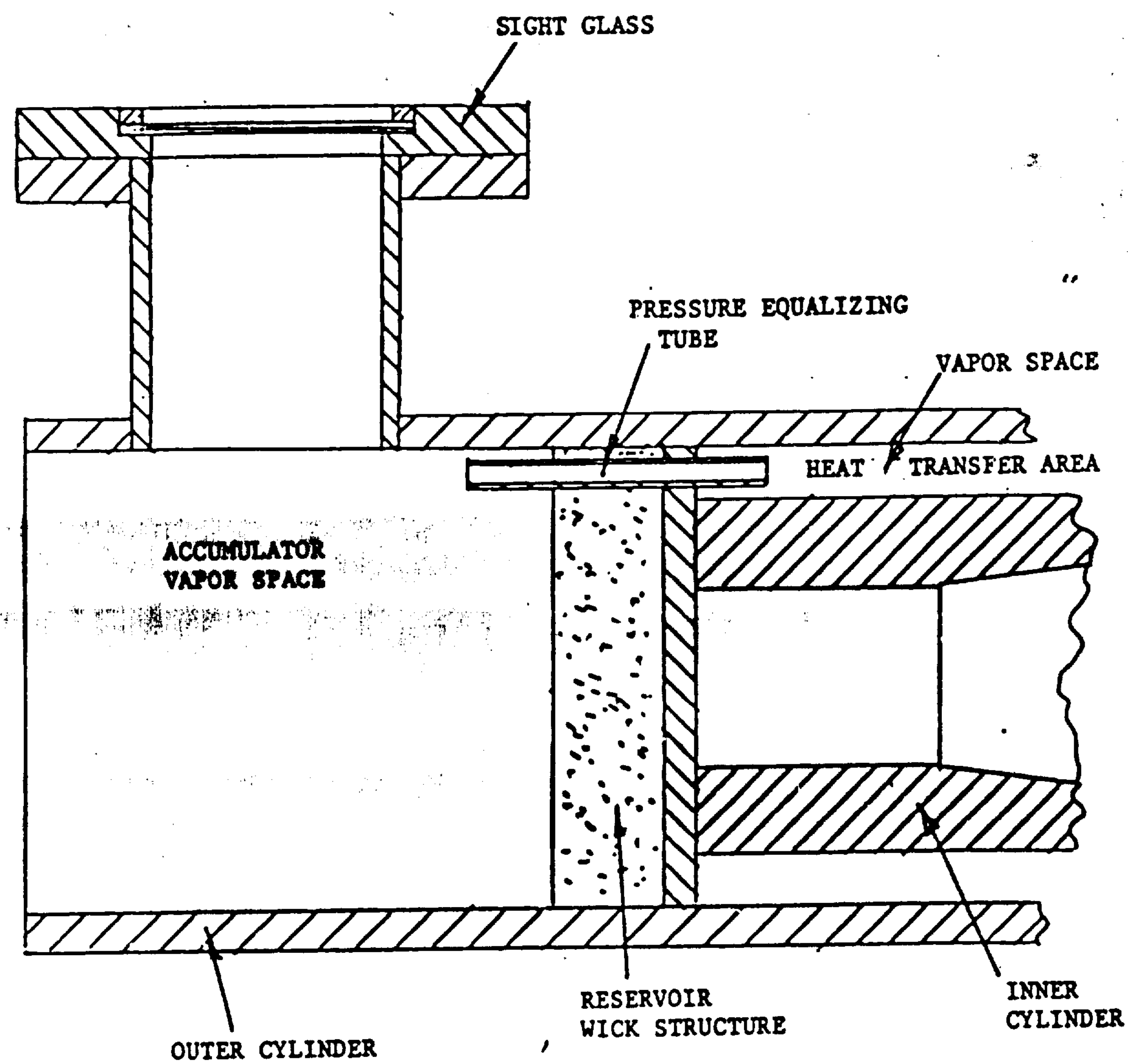


Figure 49. Thermal bus receptacle, concept II, pressure equalizing tube

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6.3 THERMAL BUS RECEPTACLE FABRICATION

Thermal bus receptacles Concept I and II were fabricated according to the design parameters listed in Table 12.

TABLE 12. Design Parameters for TBR Concept I and II

PARAMETER	TBR I (VARIABLE QUALITY VAPOR OUT)	TBR II (100% VAPOR OUT)
Diameter	7.62 cm	7.62 cm
Length	43.87 cm.	48.89
Working Fluid	Ammonia	Ammonia
Wick Structure	Axial Grooves	Powder Metal
No. Arteries	0	4
Liquid Supply Line Diameter	1.09 cm ID	1.09 cm ID
Condensate Return Line Diameter	1.57 cm ID	1.57 cm ID (vapor), 0.49 cm (liquid)
Weight	3606 grams	4909 grams

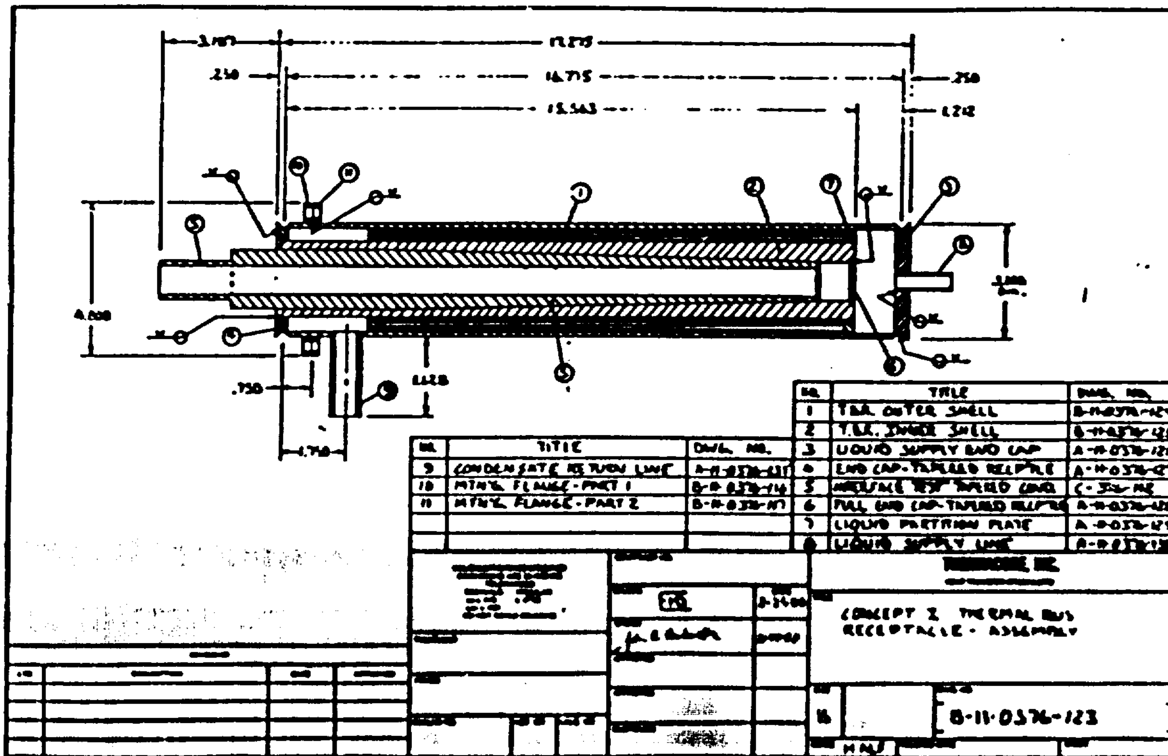
Both TBR designs were fixtured with aluminum and stainless steel mounting flanges. These flanges serve to connect the condenser of the FHPCP to the inner cylinder of the TBR. The flanges were designed with eight (8) holes, so that an even torque could be applied when assembling the components. From the tapered joint geometry designed previously, a 200 lb/in^2 contact pressure is required to obtain the lowest interface delta-T. This translates into 71 in-lbs torque per bolt for the flange.

The 100% liquid phase in - 100% vapor phase out concept has two condensate return lines. A 1.57 cm ID vapor return line is positioned 145° from the top of the TBR horizontal position as tested on earth. A 0.49 cm ID liquid return line is positioned 180° from the previously stated reference. The liquid return line is added to the 100% vapor out concept in case liquid ammonia overflows through the pressure equalizing tube into the heat transfer area. Through experimental work with the liquid level sensors and solenoid valves as supplied by the thermal bus contractor, a liquid flow sequence can be established such that this overflow should not occur.

A drawing of TBR concept I and a photograph is shown in Figure 50. A drawing of TBR Concept II is shown in Figure 51. The structural integrity of the TBR's was verified by hydrostatic testing. The hydrostatic test pressure was 1.5 times the maximum allowable working pressure of ammonia at 40°C (225 psi). A complete structural analysis of the TBR's was supplied to NASA personnel (R. Long) on April 14, 1988.

6.4 TEST MATRIX, THERMAL BUS RECEPTACLES

The performance of the thermal bus receptacles was verified by test. Each TBR was consolidated with a clamp-on cold plate and FHPCP.



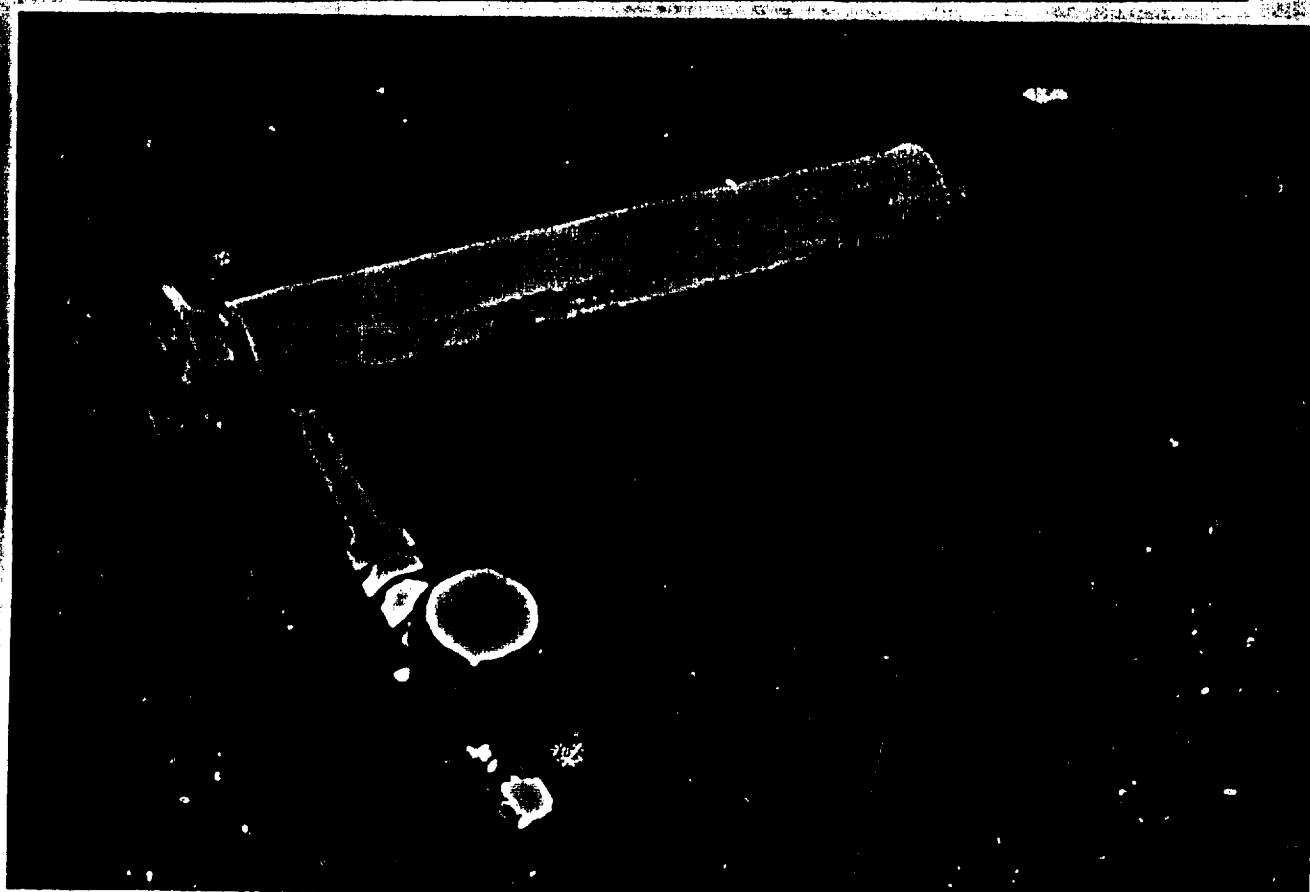


Figure 50. Thermal bus receptacle concept I assembly

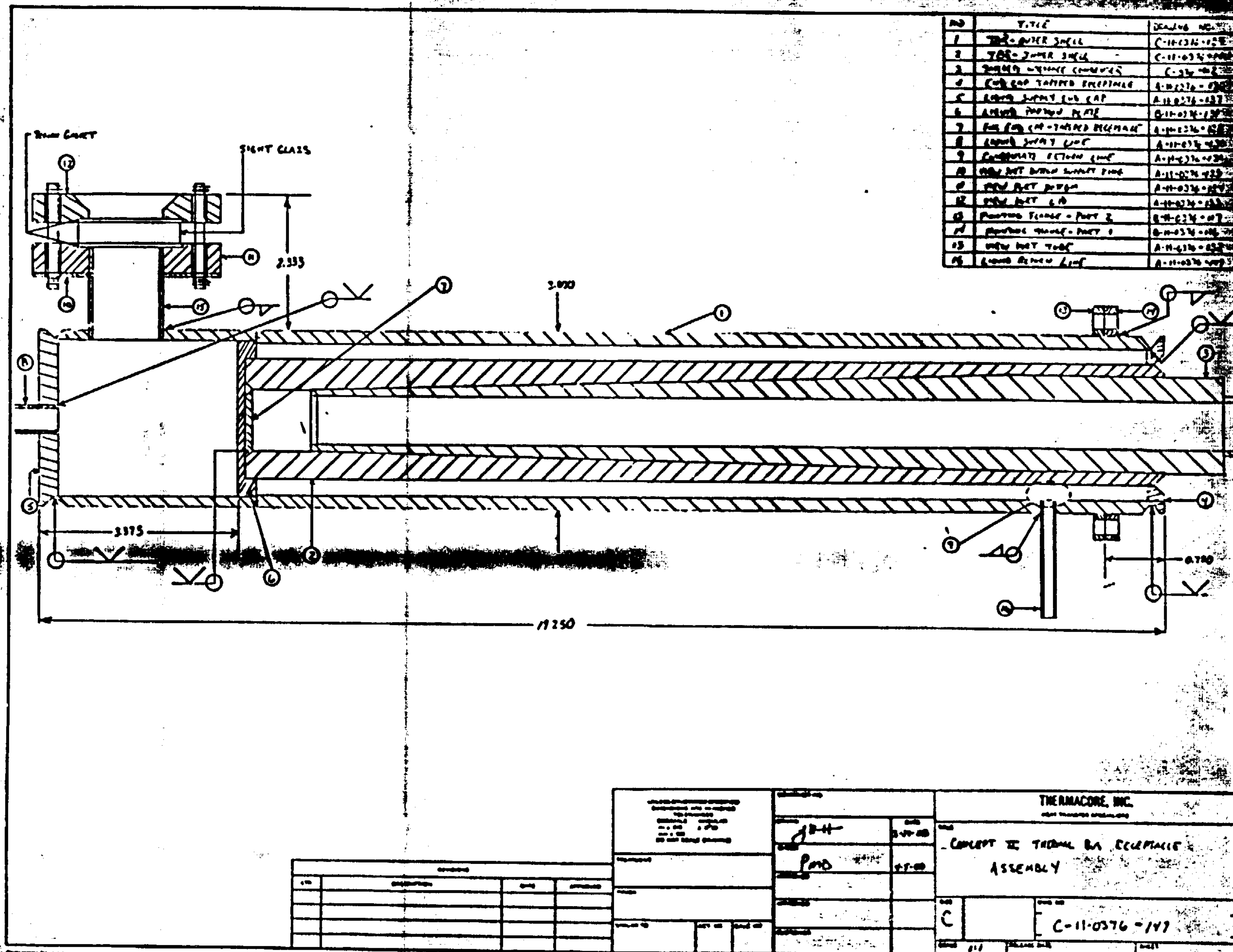


Figure 51. Thermal bus receptacle concept II assembly

6.5 TEST PROCEDURE

A simulated Space Station thermal bus was designed and fabricated to verify the design characteristics of the TBR's under load. This ammonia pumped loop used for TBR and consolidation testing utilized a magnetically-coupled gear pump to supply liquid ammonia to the TBR. The test apparatus, shown in Figures 52-53 uses an ammonia accumulator that was subcooled with a methanol/dry ice pumped loop. The vapor condenser was cooled with an ice water pumped loop. This produced a delta-T and pressure differential between the pump loop condenser and the accumulator. This pressure differential was required so that the liquid ammonia could be supplied to the pump evenly and reduce pump cavitation.

A powder metal filter was inserted in the liquid supply line before the pump to prevent any loose aluminum powder metal particles from entering the pump. The liquid return line was added to the apparatus for TBR Concept II to remove liquid that entered the heat transfer area through the pressure equalizing tube.

The entire heat transfer package which includes the hardware built in Tasks 1-5 was assembled and tested to demonstrate the integration of these modular heat transfer devices to the Space Station thermal bus. Thermal bus receptacles were inserted into a simulated Space Station thermal bus liquid supply line. A clamp-on cold plate was mounted on the outside cylinder of the TBRs and a FHPCP was inserted into the inner cylinder.

The interface medium between the clamp-on cold plate and TBR's was Emerson and Cummings TC4 thermal grease. The condenser of the FHPCP was coated with a film of diffusion pump oil to aid in retracting the condenser following the test.

Liquid ammonia was supplied to the flow through TBR (Concept I) at a constant rate. Power was applied to the plug-in or clamp on assembly until a steady state condition existed in the TBR. The data was recorded and the power was increased. Testing was terminated when the temperature of the TBR reached the pressure limitation of the liquid ammonia on the TBR wall (55°C).

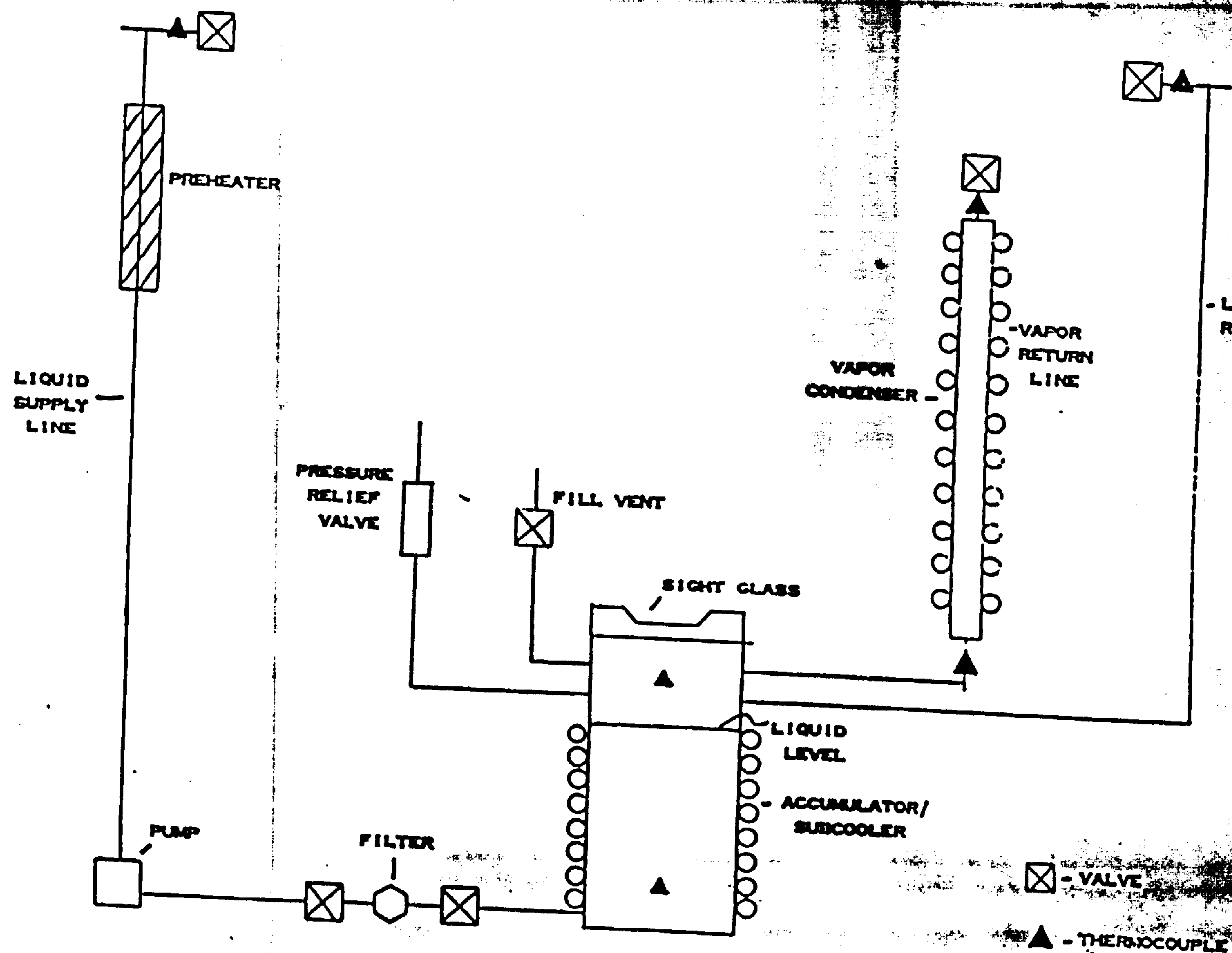


Figure 52. Representation of ammonia pumped loop

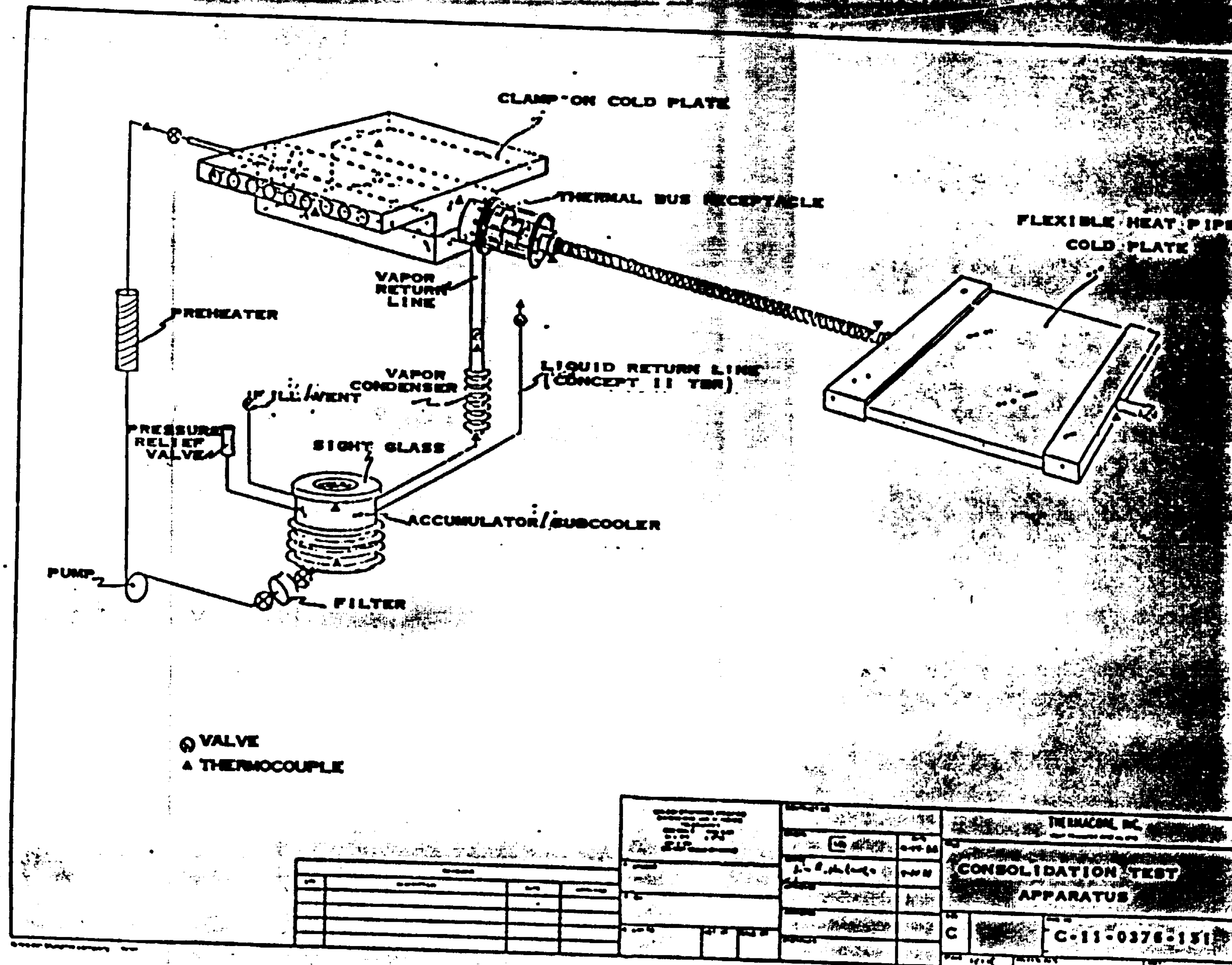


Figure 53. Consolidation test apparatus

TBR Concept II utilizes a reservoir to supply liquid ammonia to the wick structure of the heat transfer area. The level of liquid ammonia varied between two set points. The high set point was below the entrance to the pressure equalizing tube. The low set point was a distance approximately 0.635 cm from the bottom of the reservoir. As the liquid is drawn into the heat transfer area, the level would decrease. At the low set point, the valve was opened and ammonia allowed to fill the reservoir up to the high set point. The valve was then closed and the cycle was repeated until the conclusion of the test. The elapsed time between the level reaching the low set point and opening the valve was dependent upon the heat load to be rejected. A liquid level sensor installed into the reservoir would automatically open and close the valve to supply the liquid ammonia. The TBR would utilize the prime contractor's liquid level sensors.

During the accumulator filling sequence, the power was applied to the plug-in or clamp-on assembly until a steady state condition existed for the TBR. The data was recorded and the power increased. Testing was terminated when the temperature of the TBR reached the pressure limitation of the liquid ammonia on the TBR wall (55°C)

6.5 RESULTS AND CONCLUSIONS, THERMAL BUS RECEPTACLES

The test results for the performance testing of the TBR Concept I are shown in Figure 54. The figure plots power versus delta-T for the TBR flow through design. The power was applied to the TBR's by consolidating the FHPCP and clamp-on cold plates to the TBR. The performance delta-T is the temperature differential between the average FHPCP or clamp-on cold plate evaporator temperature and the TBR vapor temperature.

The plot indicates a power of 175 watts corresponding to the 10°C total delta-T limitation. The low heat transfer capability is due to a large interface delta-T between the clamp-on cold plate and the TBR and the FHPCP and the TBR. The interfaces did not match as closely as required resulting in a high interface resistance and correspondingly high delta-T. In order to achieve a low thermal resistance the interface must be machined to tighter tolerances and then highly polished.

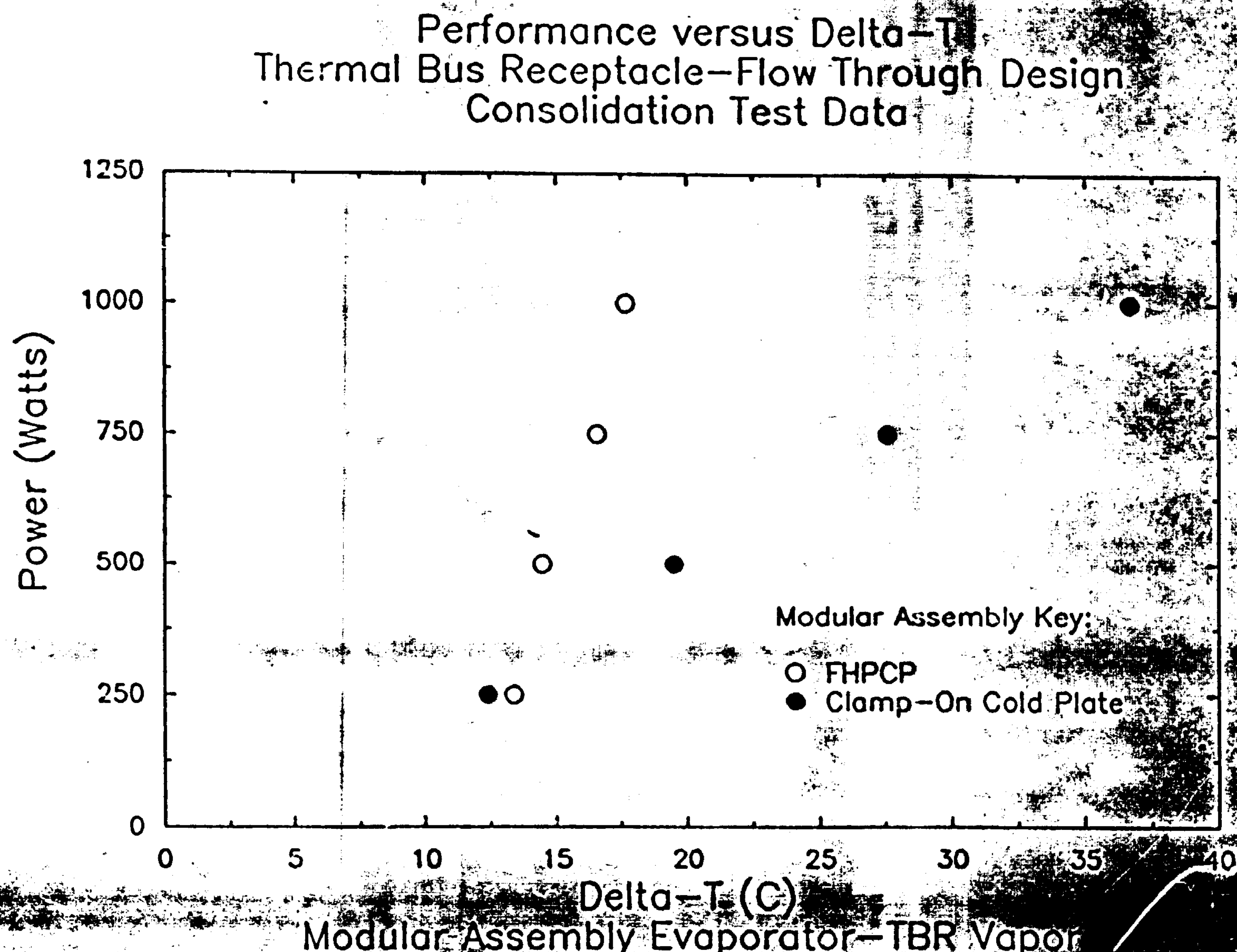


Figure 54. Performance vs. delta-T flow through design consolidation test data

The test data also show that the wick structure for both the inner and outer cylinder are capable of transferring at least 1000 watts. A photograph of consolidation testing for the flow through design is shown in Figure 55.

Test results for the reservoir TBR are shown in Figure 56. A photograph of this testing scheme is shown in Figure 57. From the data, it can be seen that the heat transfer capability of the clamp-on cold plate to the TBR and the FHPCP to the TBR is much lower than predicted. The lower than expected results are due to inadequate control over the reservoir level during testing. If the ammonia in the reservoir was allowed to drop below the given set point, the wick structure will dry out resulting in a reduction of heat transfer capability.

The clamp-on cold plate was tested individually and the results are shown in Figure 58. Clearly, the control over the liquid ammonia level in the reservoir is improved. A 750 watt heat transfer capability was achieved corresponding to the 10°C delta-T limitation. This heat transfer capability/delta-T limitation is in agreement with the performance data for the 30 cm x 30 cm ammonia clamp-on cold plate tested in Task 3. These data also show a low interface resistance between the mating components.

Individually testing the FHPCP did not yield any improvement in performance over the data presented in Figure 56. Analysis of the data indicates that a large interface delta-T was present between the mating components. Again, this would cause a drop in heat transfer capability between the FHPCP and TBR.

The FHPCP was removed from the TBR and a tapered condenser was bolted into place. The tapered condenser was instrumented with a 1.27 cm, 2000 watt cartridge heater which was inserted down the center of the condenser. The tapered condenser/cartridge heater arrangement was inserted to determine if a lower thermal resistance was achievable with a different condenser than that of the FHPCP. Figure 59 shows the performance data from this test. It is evident that the condenser surface finish better matched that of the surface finish of the the FHPCP condenser. The data also suggest that the TBR with a reservoir design is

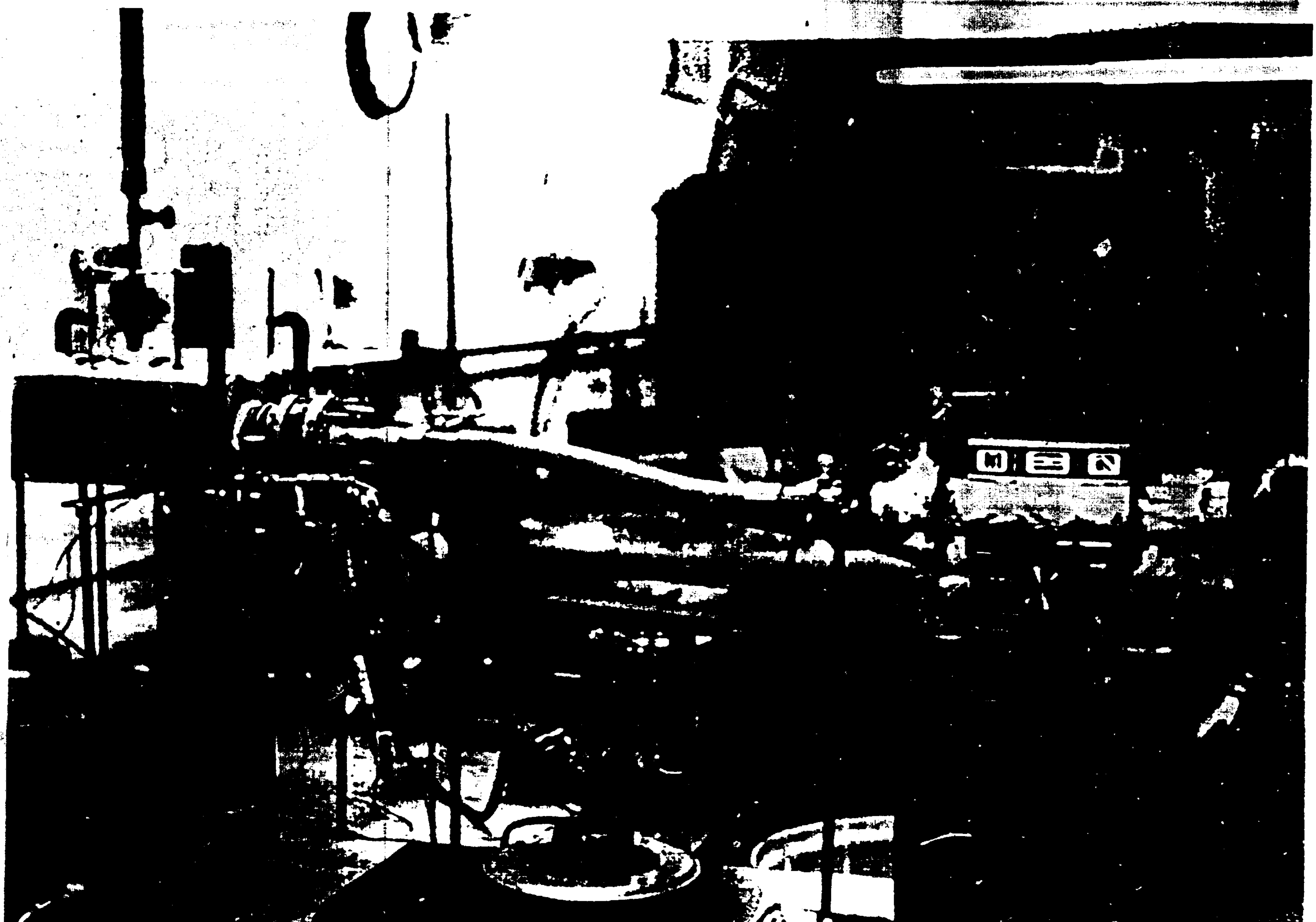


Figure 55. Consolidation testing of the heat transfer system using the flow through

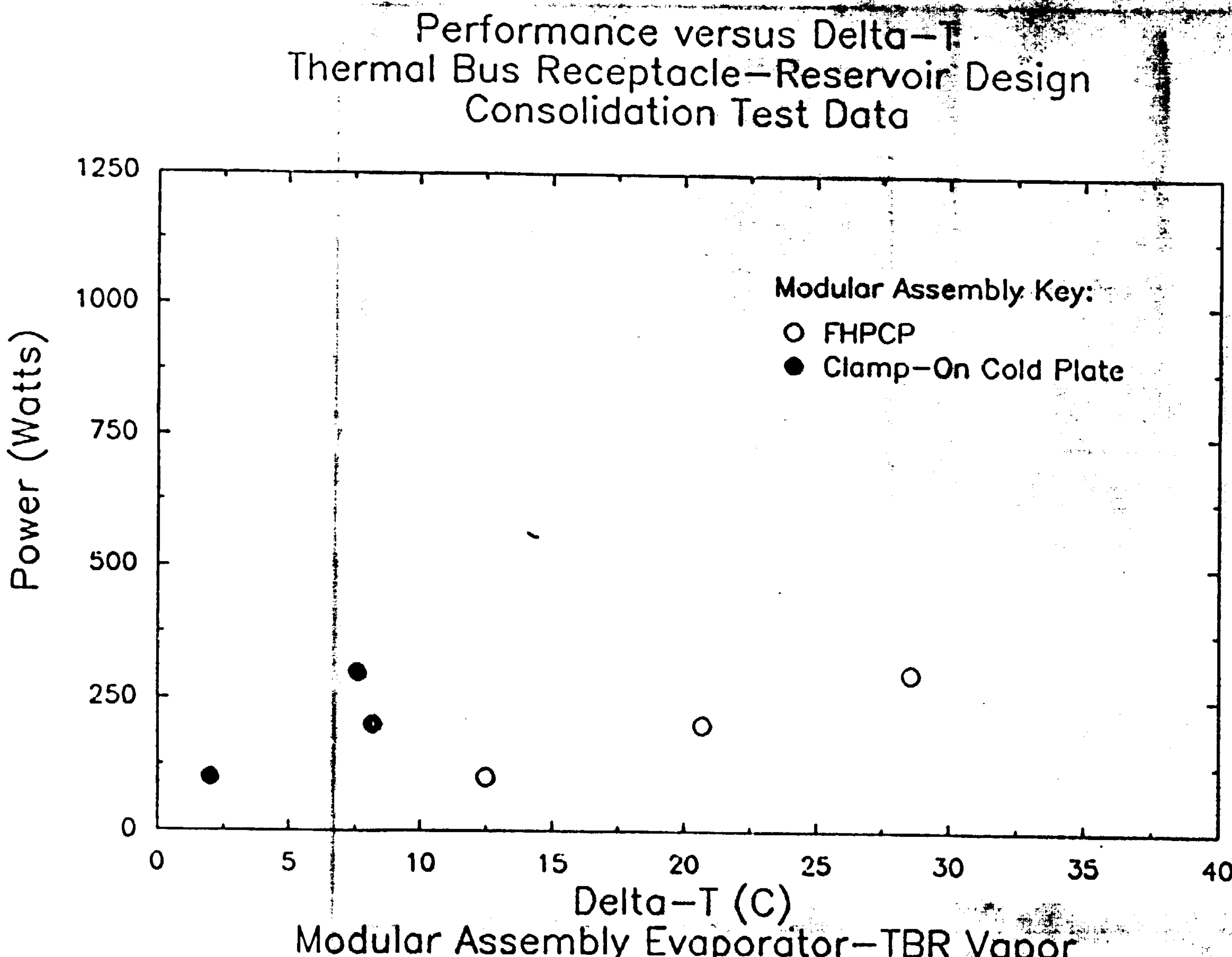


Figure 56. Performance vs. delta-T thermal bus receptacle - reservoir design consolidation test data

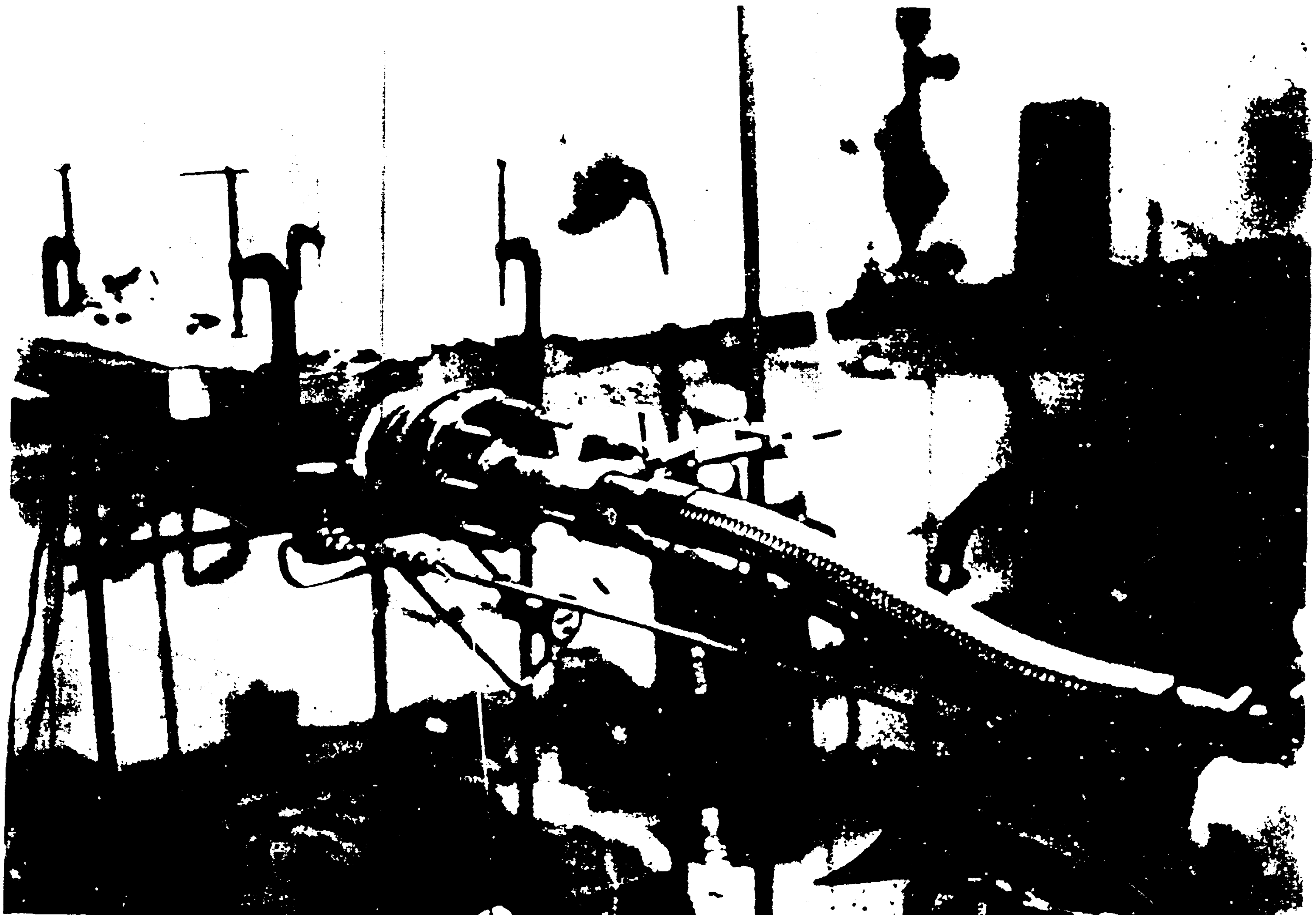


Figure 57. Consolidation of the heat transfer package using the reservoir design TBR

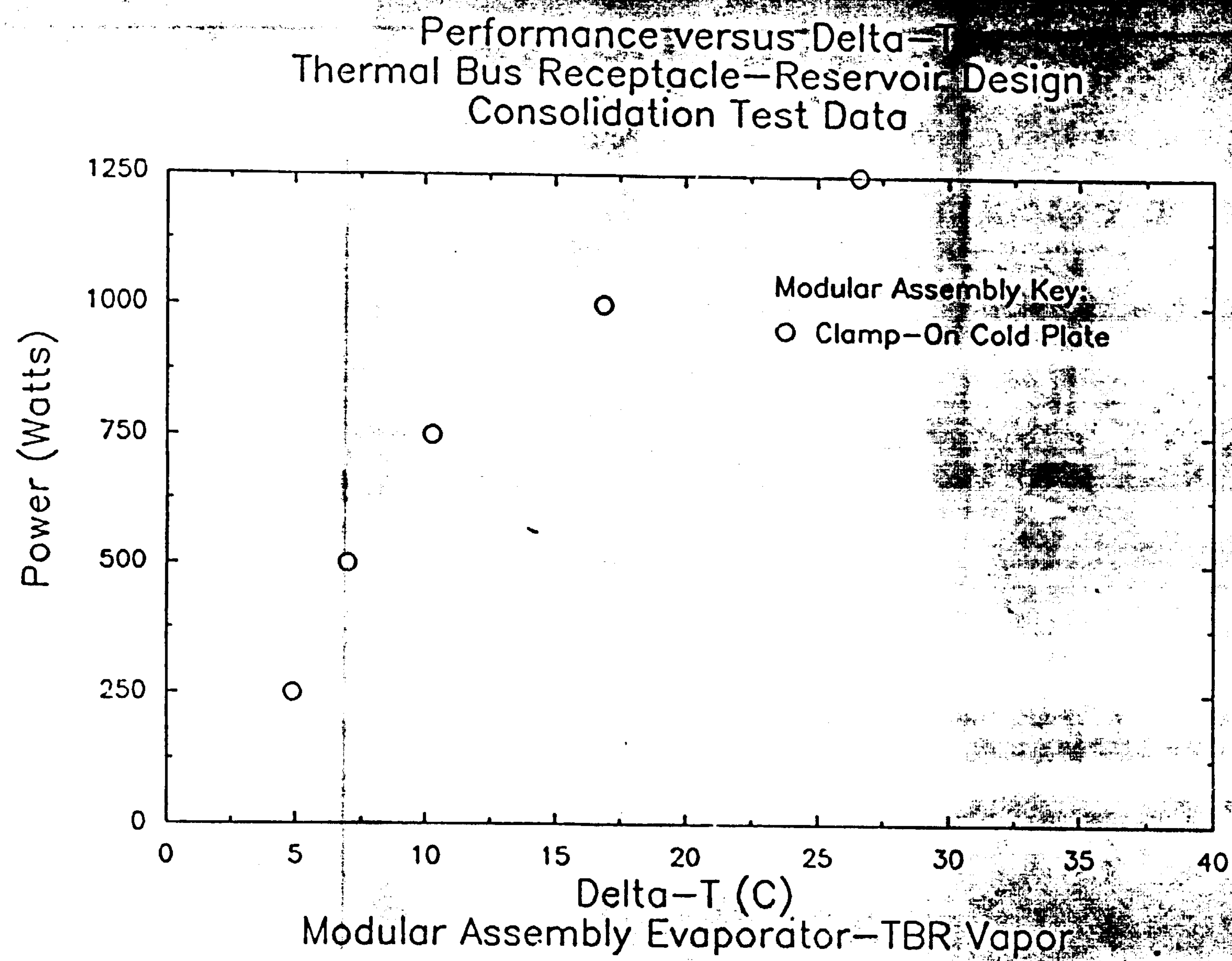


Figure 58. <u>Performance vs. delta-T thermal bus receptacle - reservoir design consolidation test data</u>

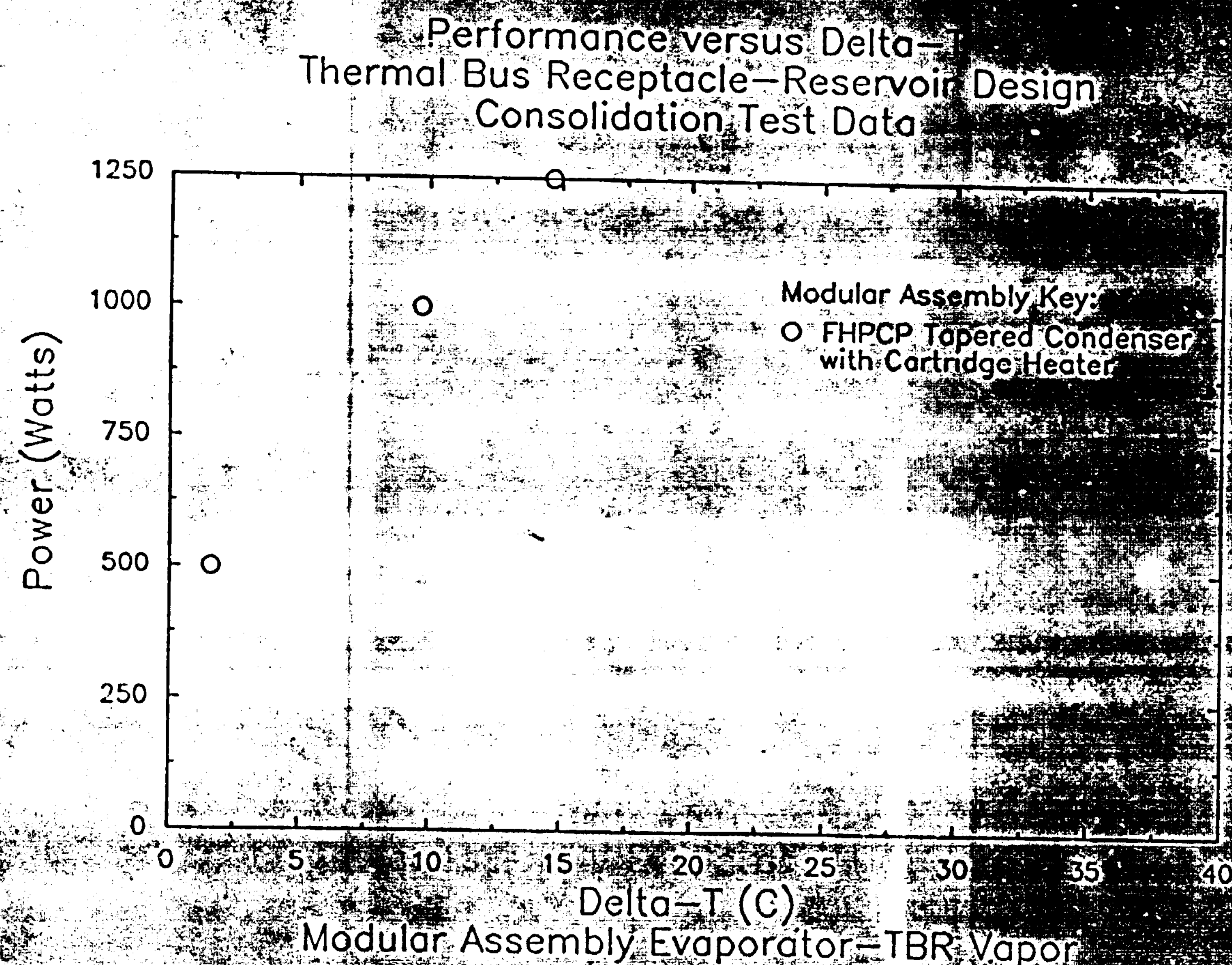


Figure 59. Performance vs. delta-T thermal bus receptacle - reservoir design consolidation test data

capable of transferring at least 1250 W from the clamp-on or plug-in assemblies, as long as there is strict control over the ammonia fluid le"el in the reservoir.

6.6 CONCLUSIONS: THERMAL BUS RECEPTACLE

- Both TBR designs are capable of transferring the waste heat from the clamp-on or plug-in assemblies to the thermal bus.

- Strict control over the ammonia liquid level in the reservoir of TBR II is essential to provide uniform heat rejection.

- Closely matched surfaces and surface finishes are required between mating components in order to reduce the interface resistance and delta-T.

6.7 RECOMMENDATIONS: THERMAL BUS RECEPTACLE

- Future TBR designs with a reservoir should include a liquid level sensor and solenoid valve to provide tighter control over the liquid ammonia level.

- Future fabrication of the heat transfer package (clamp-on cold plate, flexible heat pipe cold plate and thermal bus receptacle) should include a procedure to conduct a final machining and polishing of the mating components after component fabrication.

7.0 CONSOLIDATION OF HEAT ACQUISITION-HEAT TRANSFER SYSTEM

The objective of this task was to assemble the hardware built and tested individually and to consolidate the hardware to demonstrate a total heat transfer package. The demonstration of the hardware consolidation was successful. Two thermal bus receptacles were designed to accommodate the clamp-on cold plate and flexible heat pipe cold plate integration. The requirements for the total heat transfer package are shown below.

7.1 REQUIREMENTS, CONSOLIDATION

The requirements for the total heat transfer consolidation are shown in Table 13.

TABLE 13. Consolidation of Heat Transfer Package Requirements

PARAMETER	CLAMP-ON COLD PLATE	FLEXIBLE HEAT PIPE COLD PLATE	THERMAL BUS RECEPTACLE
Mass	Minimize	Minimize	Minimize
Operating Temperature (°C)	0 - 40	0 - 40	0 - 40
Dimensions (cm)	30 x 30 30 x 60 30 x 90	30 x 30, 2.54 dia holes 30 x 30, 1.58 dia. holes	7.62 diameter
Temperature Drop (°C)	8 - 10	8 - 10	---
Total Power (W)	2000 - 3000	>1000	Maximize

7.2 RESULTS AND CONCLUSIONS, CONSOLIDATION

The total heat transfer package was consolidated and tested in Task 6 in order to verify the performance of the thermal bus receptacles. Consolidation testing with the flow through TBR yielded a 175 watt heat transfer capability which was limited by the 10°C maximum delta-T requirement. The low heat transfer capability was caused by a large interface delta-T between the clamp-on cold plate and the TBR and the FHPCP and the TBR.

Consolidation testing using the TBR with a reservoir yielded a 100 watt heat transfer capability with the FHPCP and a 250 watt capability with the clamp-on cold plate. The test was terminated because the clamp-on cold plate temperature approached the 55°C temperature limitation. The low performance was due to inadequate control over the reservoir level during testing. If the ammonia in the reservoir falls below the minimum set-point, the wick structure will dry out and reduce the heat transfer capability.

Individual testing of the FHPCP and clamp-on cold plate with the reservoir TBR was conducted to determine if better control over the ammonia level was achievable. Individual testing of the clamp-on cold plate resulted in a 750 watt heat transfer capability corresponding to the 10°C maximum delta-T limitation.

Individual testing of the FHPCP did not improve the test results. A large thermal resistance between the FHPCP tapered condenser and TBR was the cause of the low heat transfer capability. The FHPCP was removed and a tapered condenser only was inserted into the reservoir TBR. The condenser was instrumented with a 1.27 cm diameter, 2000 watt cartridge heater. Testing yielded a 1000 watt heat transfer capability which corresponds to a 10°C maximum delta-T limitation. It is apparent that the surface finishes of the mating components matched well resulting in a lower interface delta-T.

The results from the consolidation tests are summarized in Table 14.

TABLE 14. Consolidation Test Results

MODULAR ASSEMBLY	HEAT TRANSFER CORRESPONDING TO THE 10°C DELTA-T LIMITATION (W) TBR I (FLOW THROUGH)	TBR II (RESERVOIR)
Clamp-On Cold Plate, FHPCP	175	100 W (FHPCP), 250 (COCP)
Clamp-On Cold Plate	-	750
Tapered Condenser	-	1000

7.2.1 Flow Through Cold Plate

As an additional task, Thermacore combined the technology developments of the clamp-on cold plate, FHPCP and TBR assembly and fabricated a flow through cold plate. The flow through cold plate uses the wick structure developed for the clamp-on cold plate; the artery and vapor core manifold developed for the FHPCP; and the variable quality vapor out characteristics from the flow through TBR.

The flow through cold plate shown in Figure 60 is assembled directly into the Space Station thermal bus similar to the TBR. The liquid supply line is integrated directly into the artery manifold where the liquid ammonia is supplied to the powder metal wick. The power is input on the top of the cold plate. The heat input is transferred into the Space Station radiators through two return lines; one for vapor and one for liquid. The result is a high performance, lightweight heat transfer device to reject waste heat from equipment and experiments.

Test data indicates the flow through cold plate carried 3000 watts with a <1°C delta-T. The test was terminated when the heater capacity was reached.

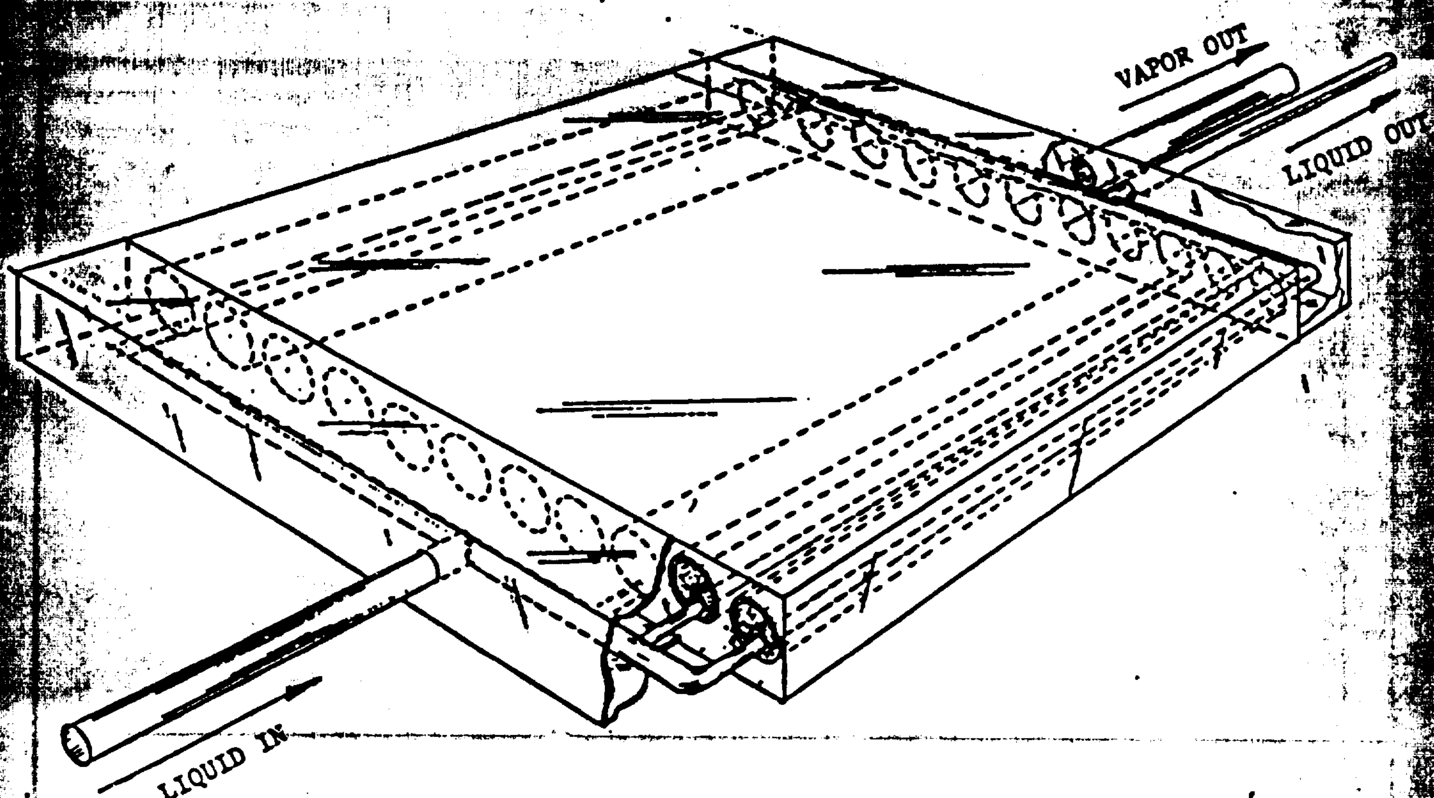


Figure 60. Flow through cold plate design

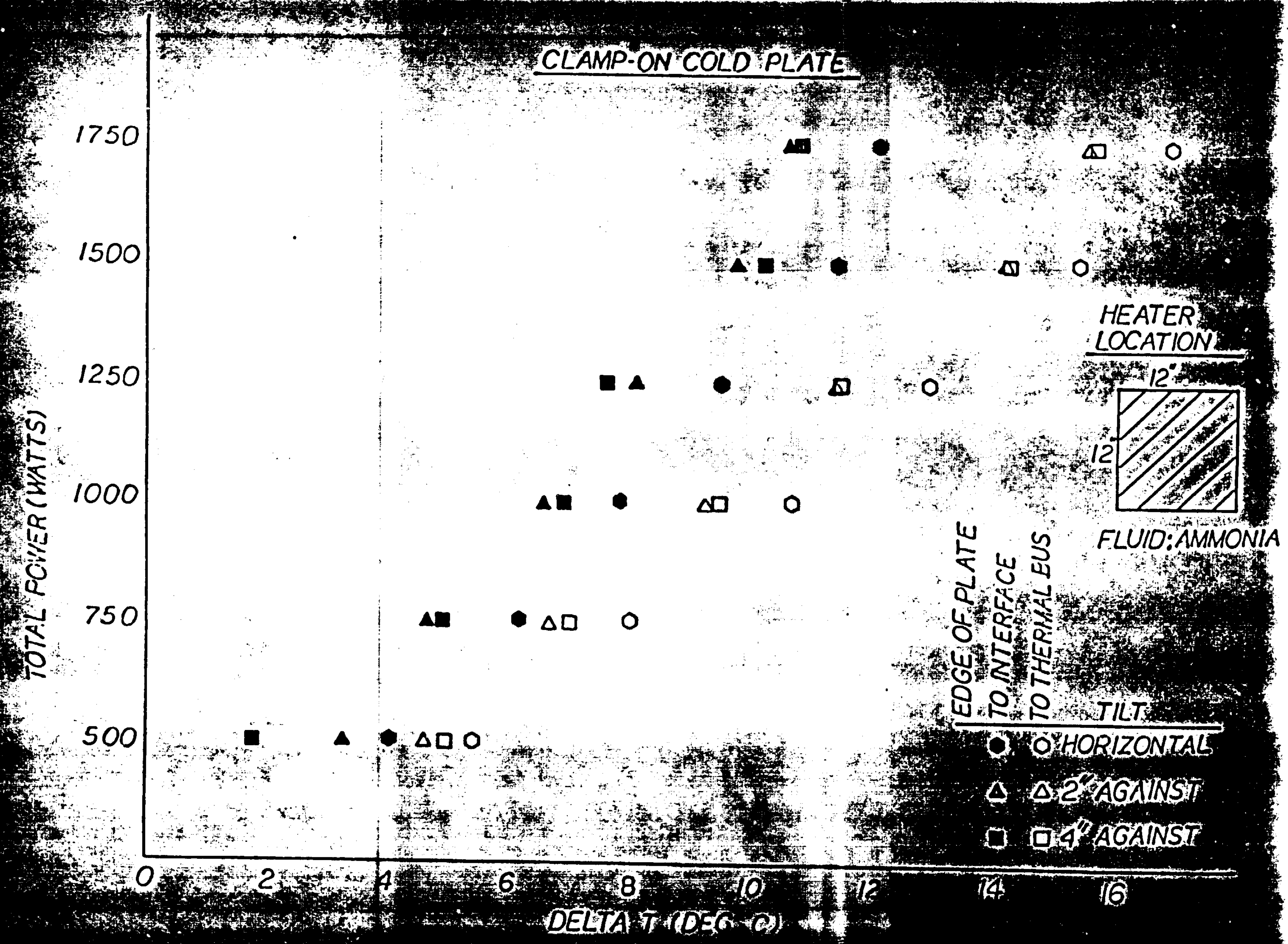
7.2.1 Conclusions: Consolidation

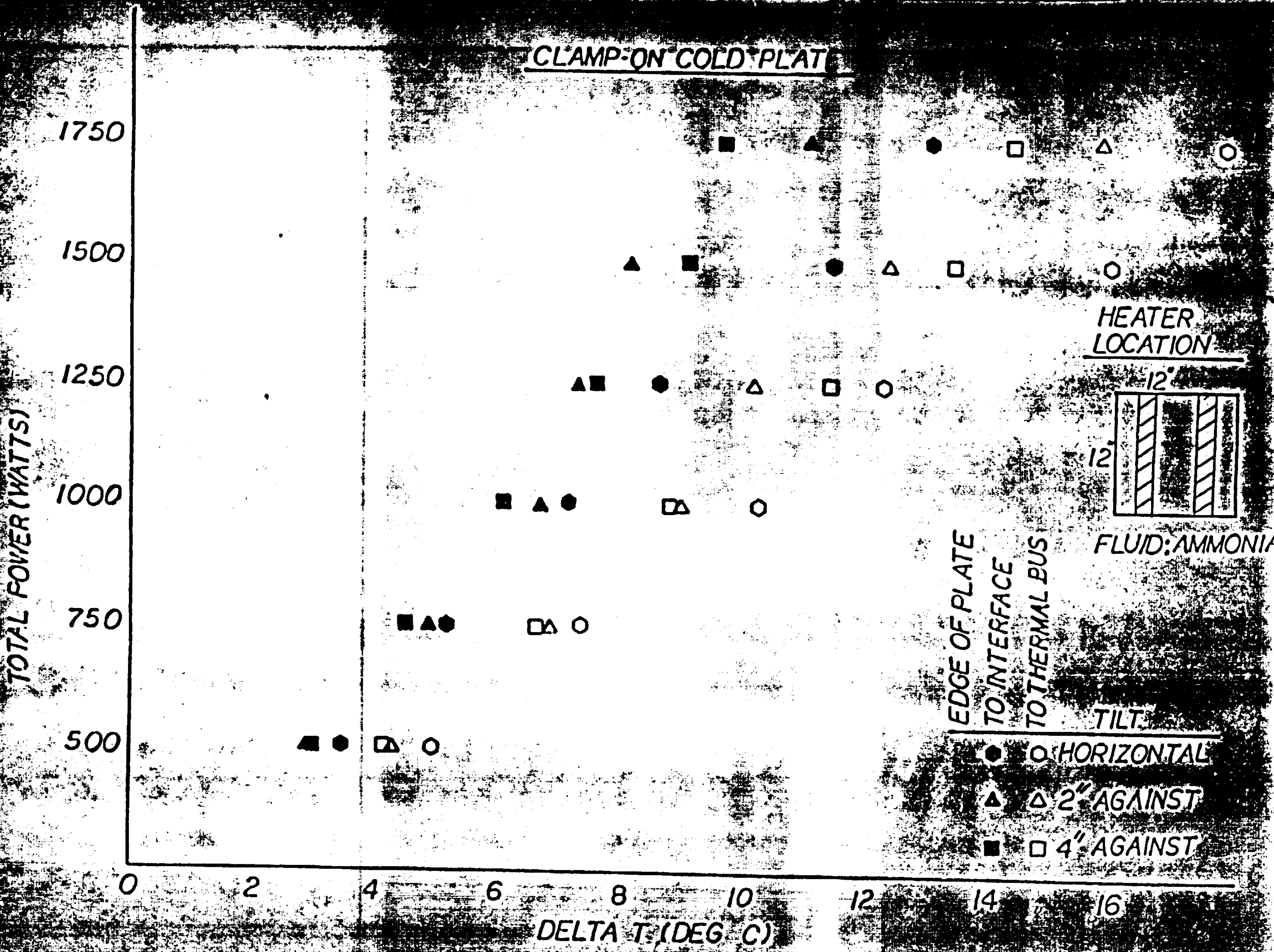
- The flow through TBR and the TBR with a reservoir are capable of transferring at least 1000 watts to the Space Station radiators.
- The interface between the FHPCP condenser and the flow through TBR did not mechanically match well enough to achieve a low interface delta-T. The results is a low heat transfer capability.
- The flow through cold plate is capable of carrying at least 3000 watts with a <1°C delta-T.
- The flow through TBR weighed 3604 grams. The TBR with a reservoir weighed 4907 grams.

7.2.2 Recommendations: Consolidation

- Future consolidation designs should include a procedure to conduct a final machining and polishing of the mating components after fabrication.
- The TBR with a reservoir should be designed to include the Space Station thermal bus liquid level sensors and valves in order to provide tighter control over the liquid ammonia level.
- Continue to investigate materials and designs to reduce the weight of the modular assemblies.
- Continue the development of the flow through cold plate for integration into the Space Station thermal bus.

APPENDIX A


CLAMP-ON COLD PLATE
TOTAL POWER (WATTS)
1750
1500
1250
1000
750
500
0
2
4
6
8
10
12
14
16
DELTA T (DEG C)
HEATER LOCATION
12"
12
FLUID: AMMONIA
EDGE OF PLATE TO INTERFACE
TO THERMAL BUS
TILT
HORIZONTAL
2" AGAINST
4" AGAINST



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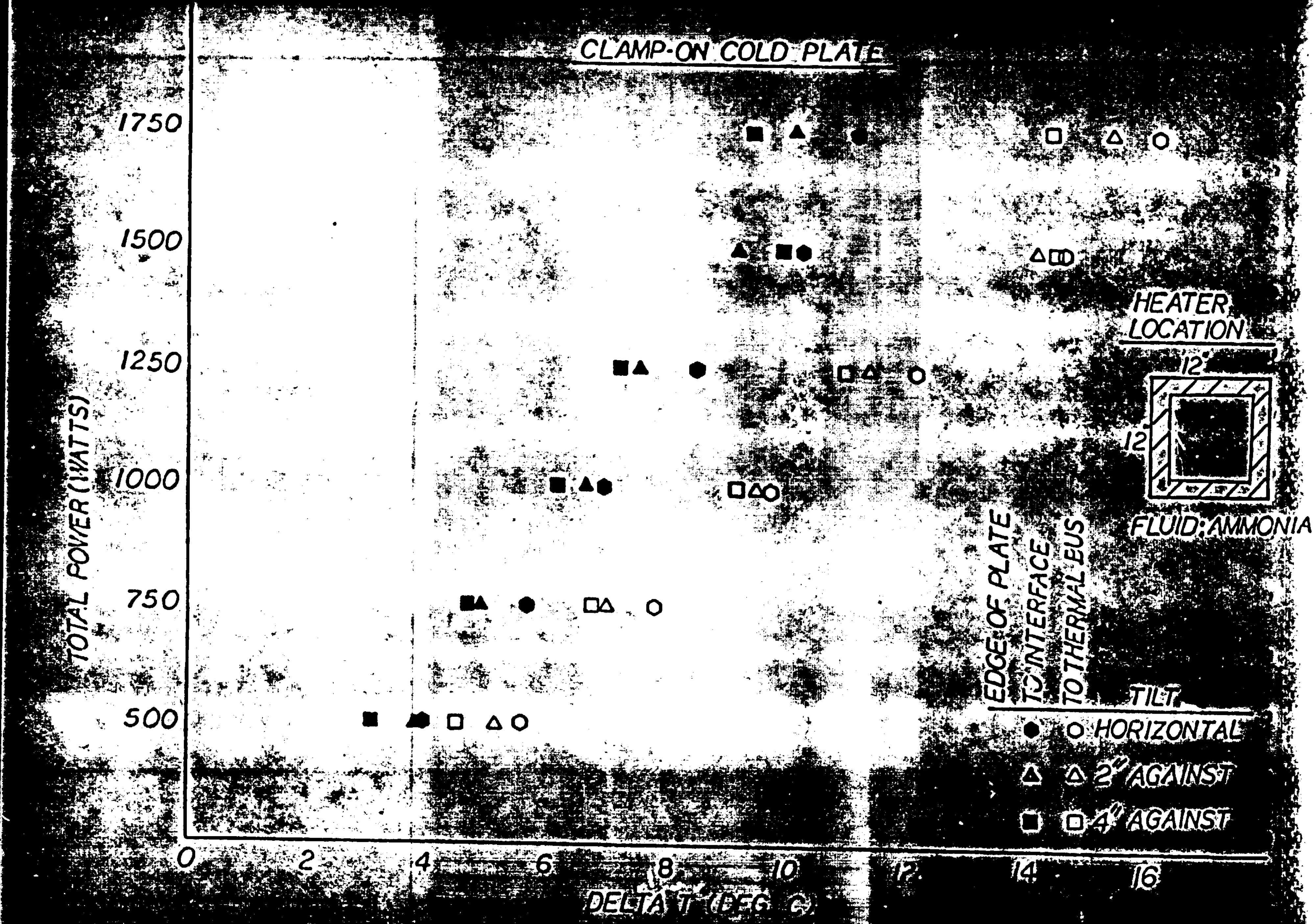

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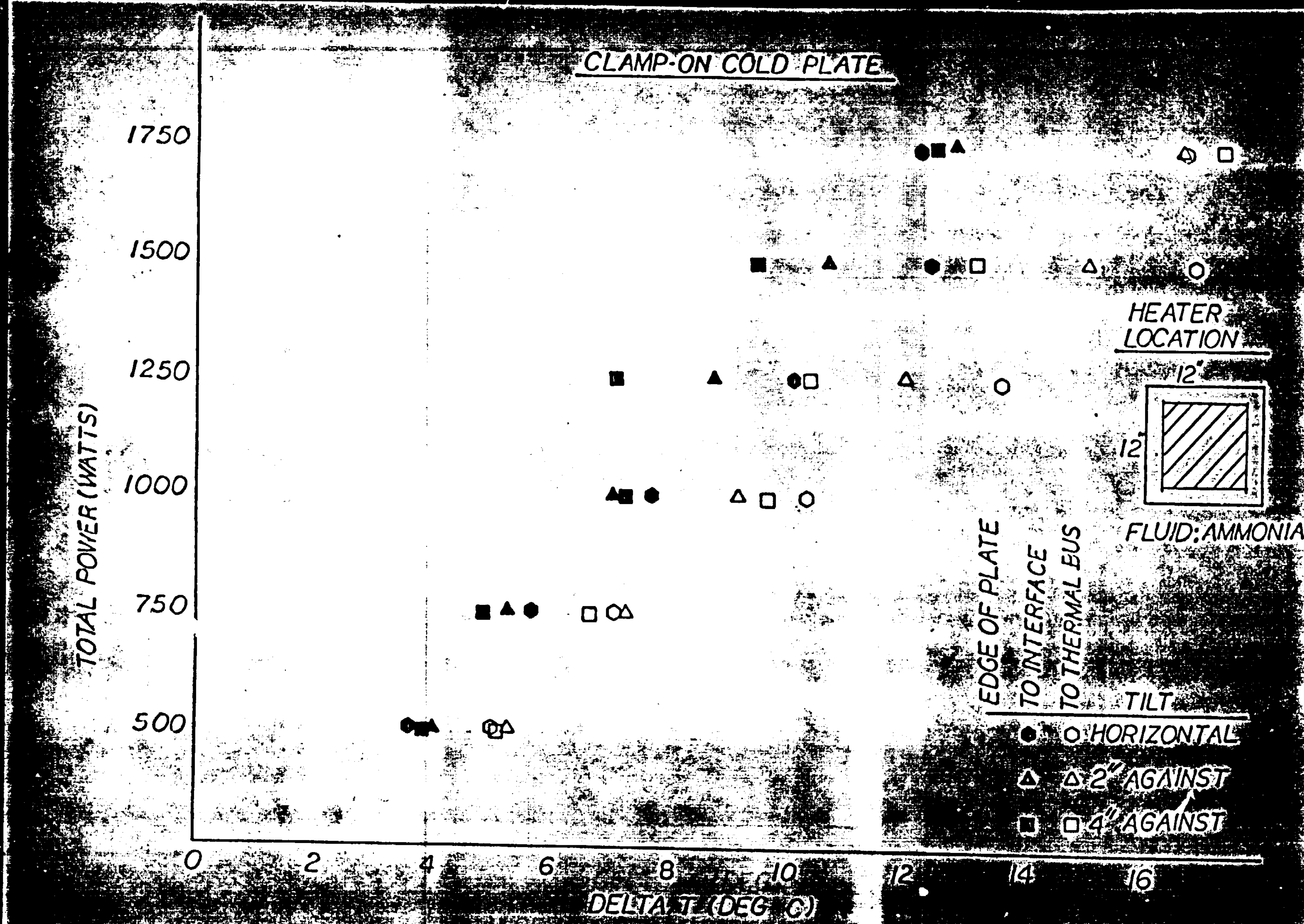

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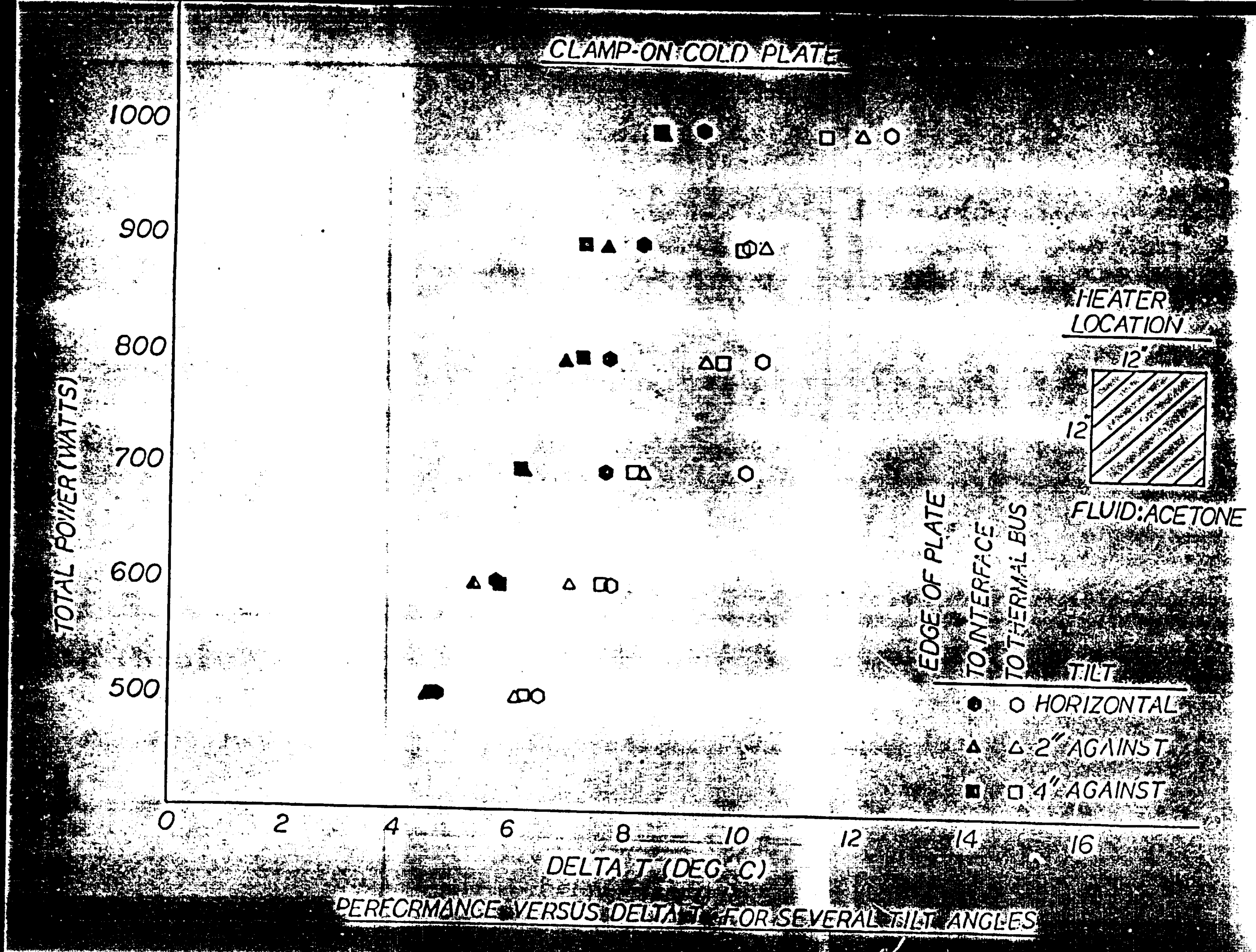
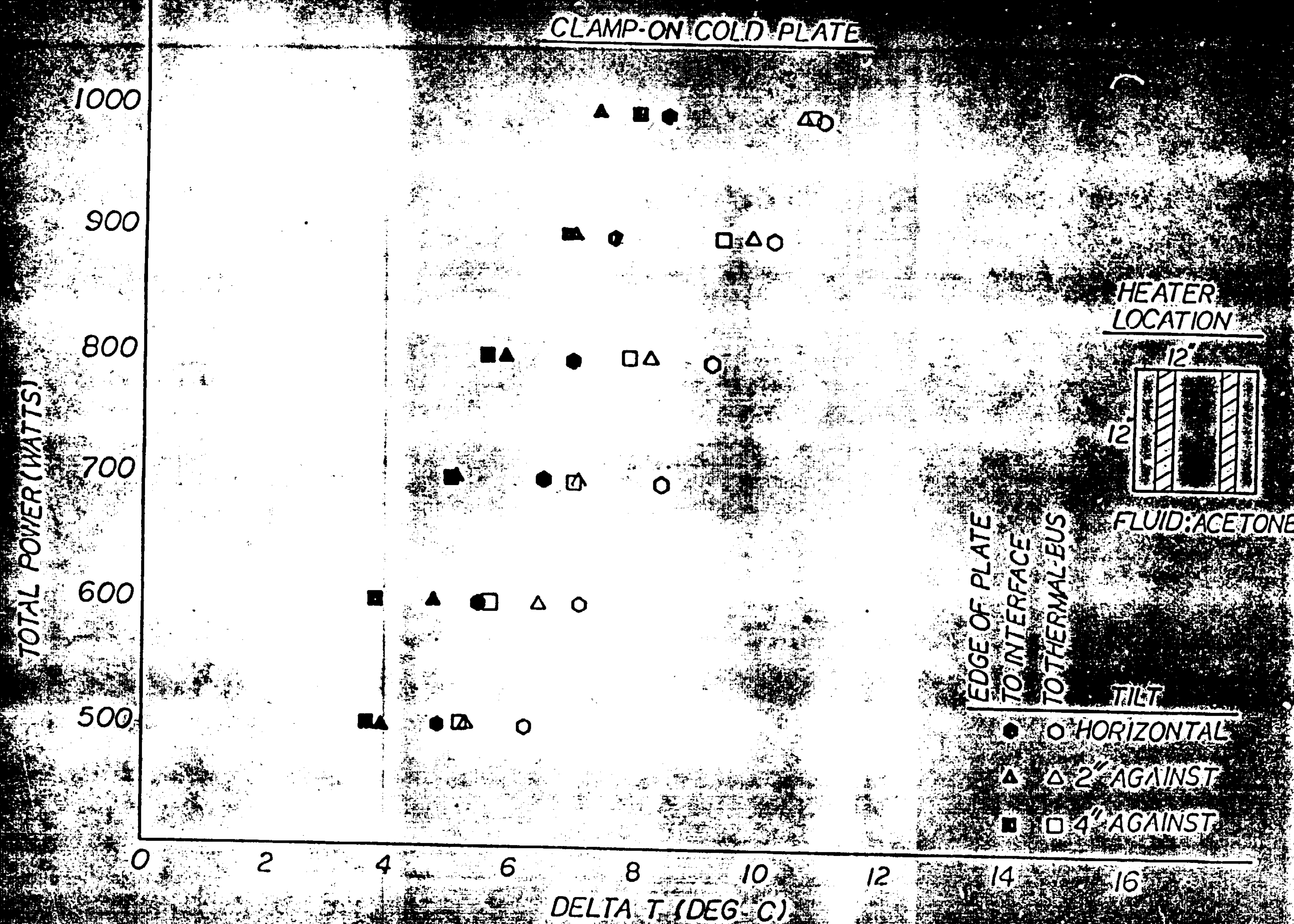


CLAMP-ON COLD PLATE
TOTAL POWER (WATTS)
1000
900
800
700
600
500
0
2
4
6
8
10
12
14
16
DELTA T (DEG C)
HEATER LOCATION
12"
12"
FLUID: ACETONE
EDGE OF PLATE
TO INTERFACE
TO THERMAL BUS
TILT
HORIZONTAL
2" AGAINST
4" AGAINST
PERFORMANCE VERSUS DELTA T FOR SEVERAL TILT ANGLES



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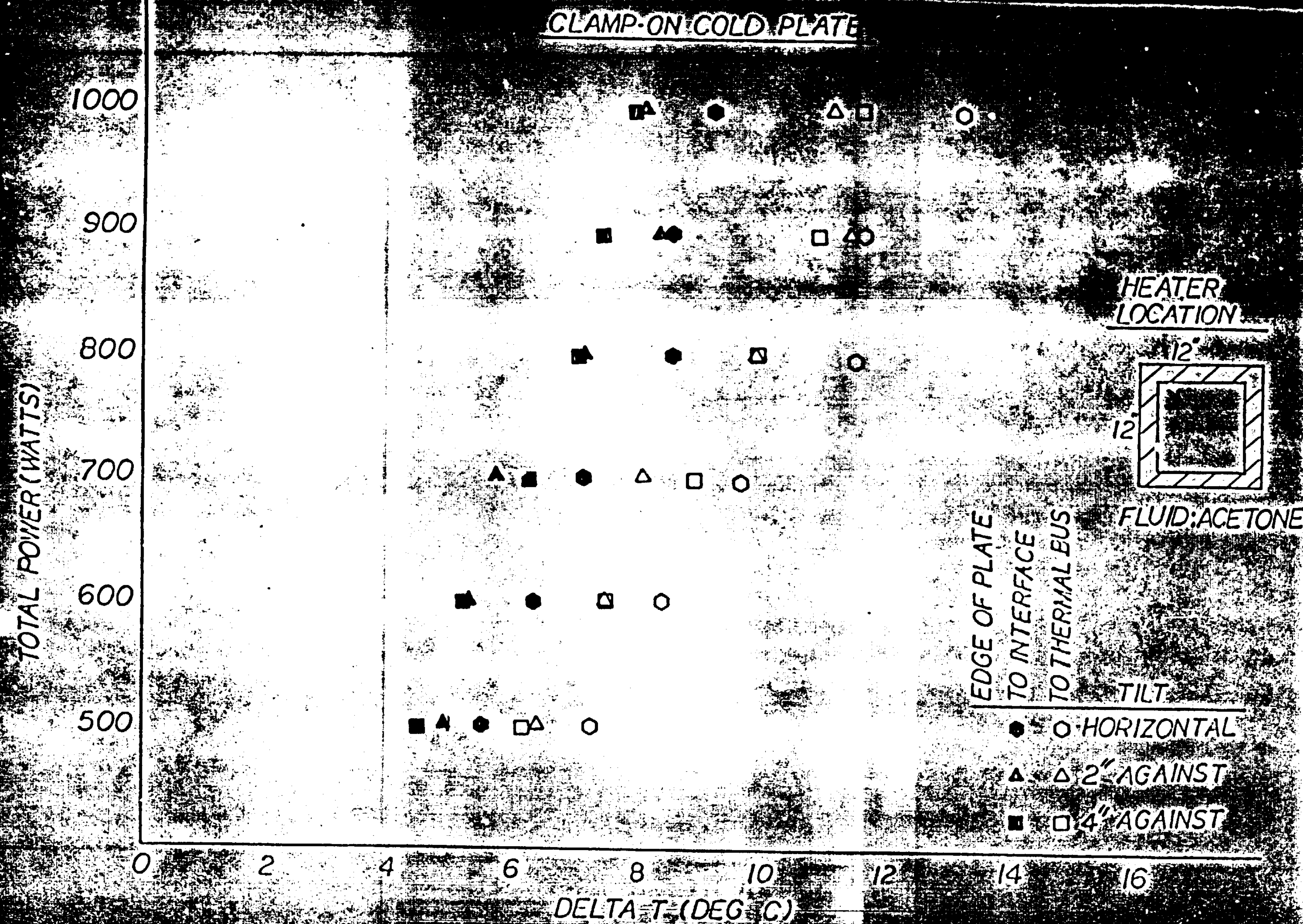

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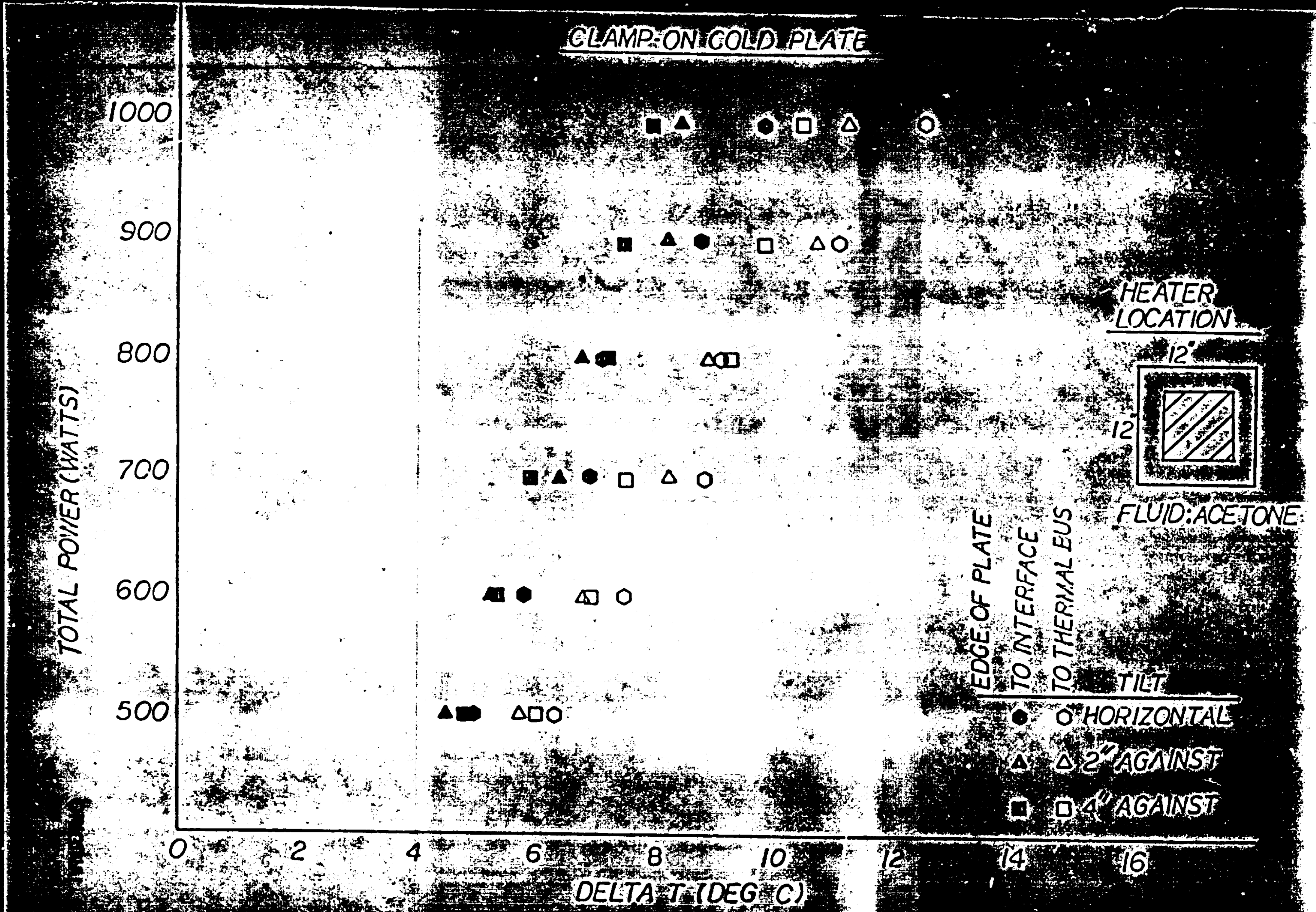

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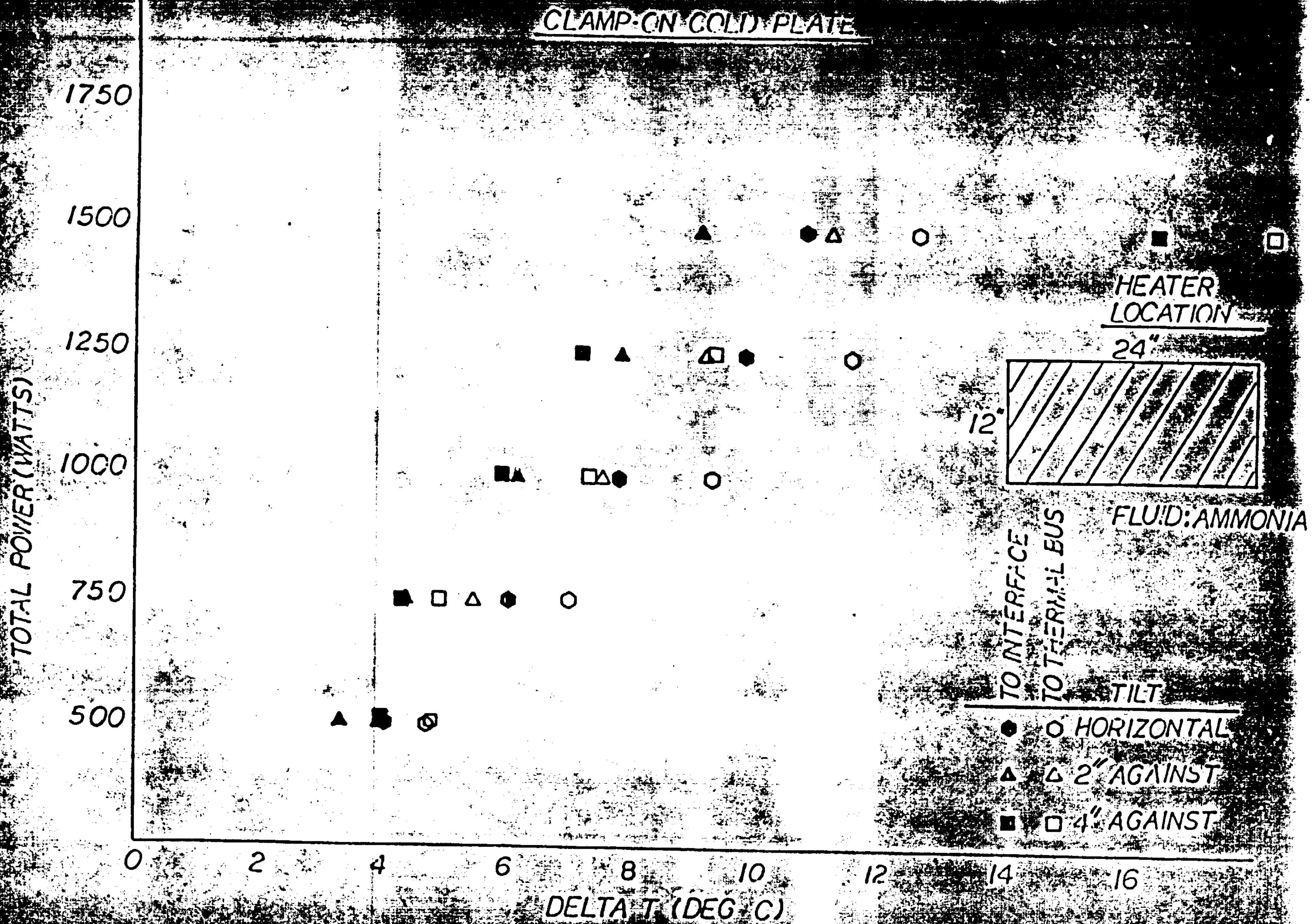

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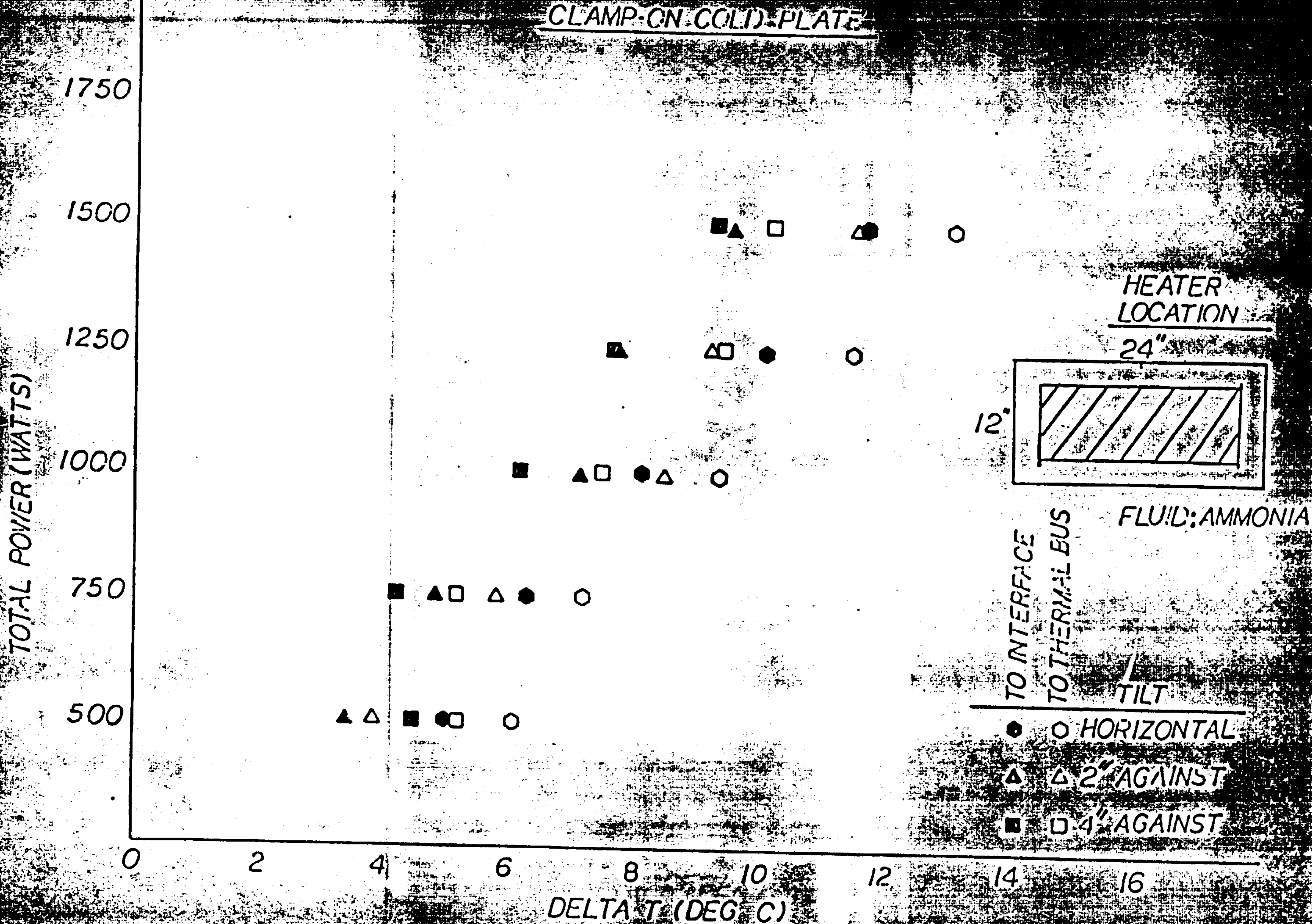

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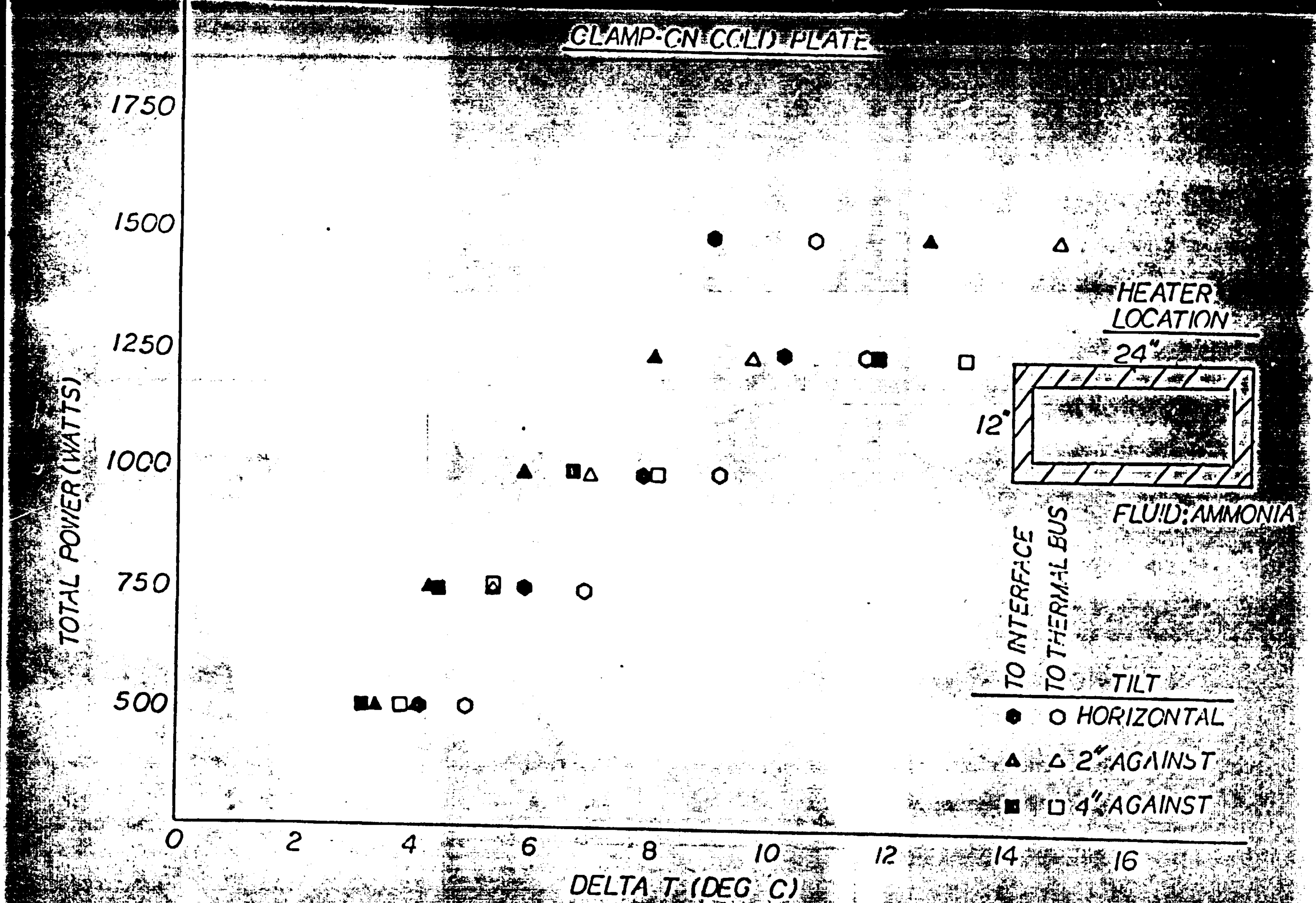

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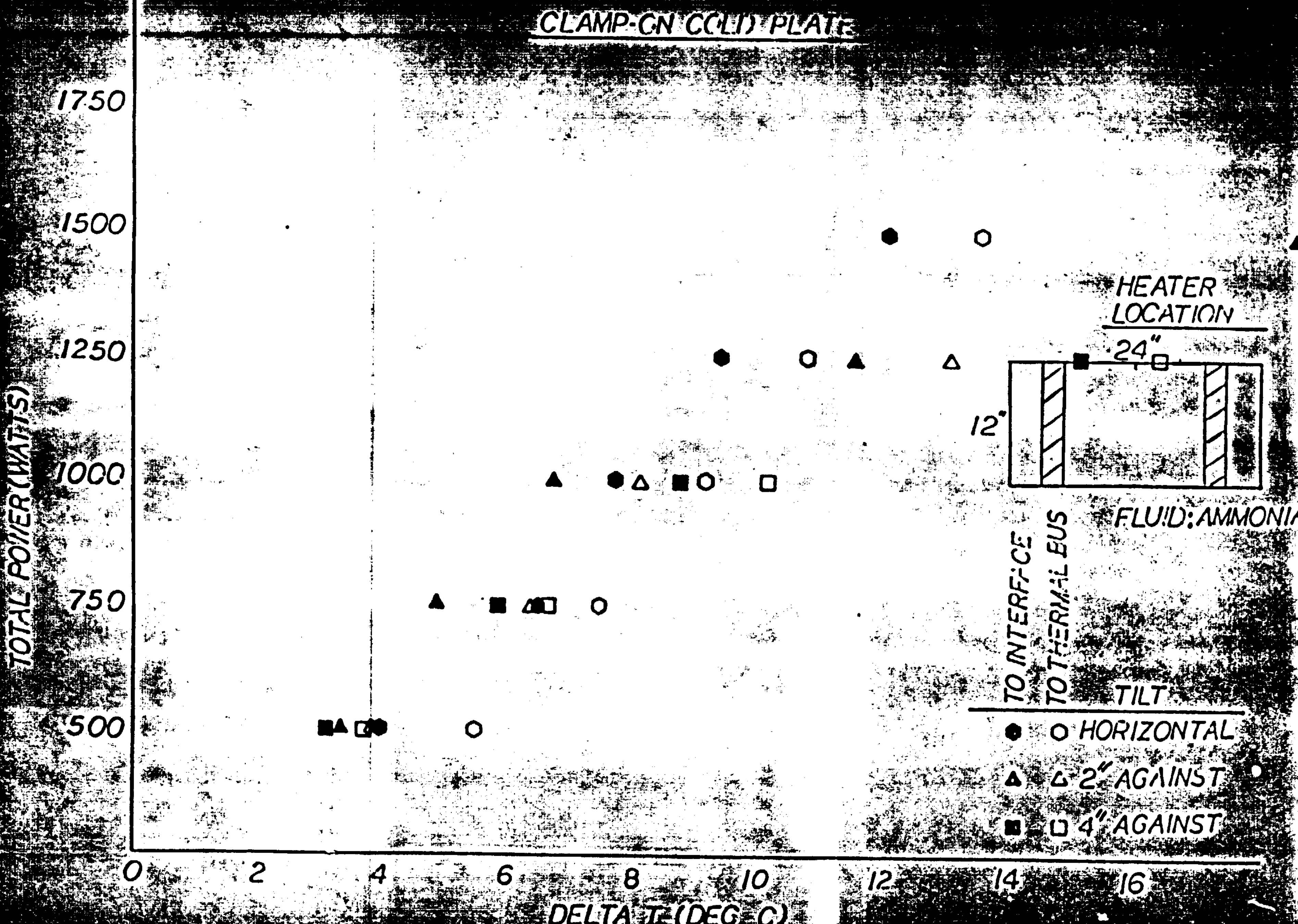

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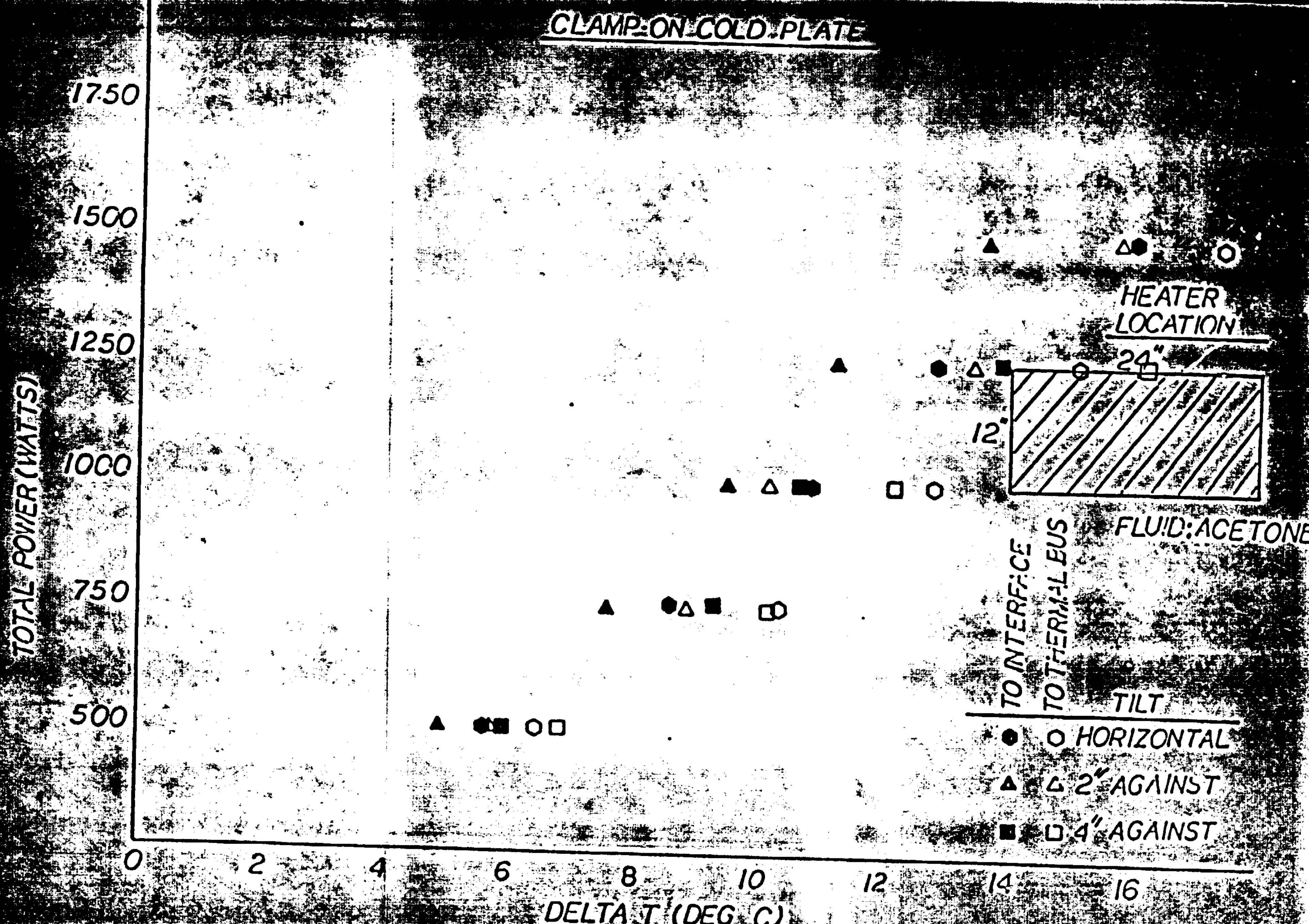

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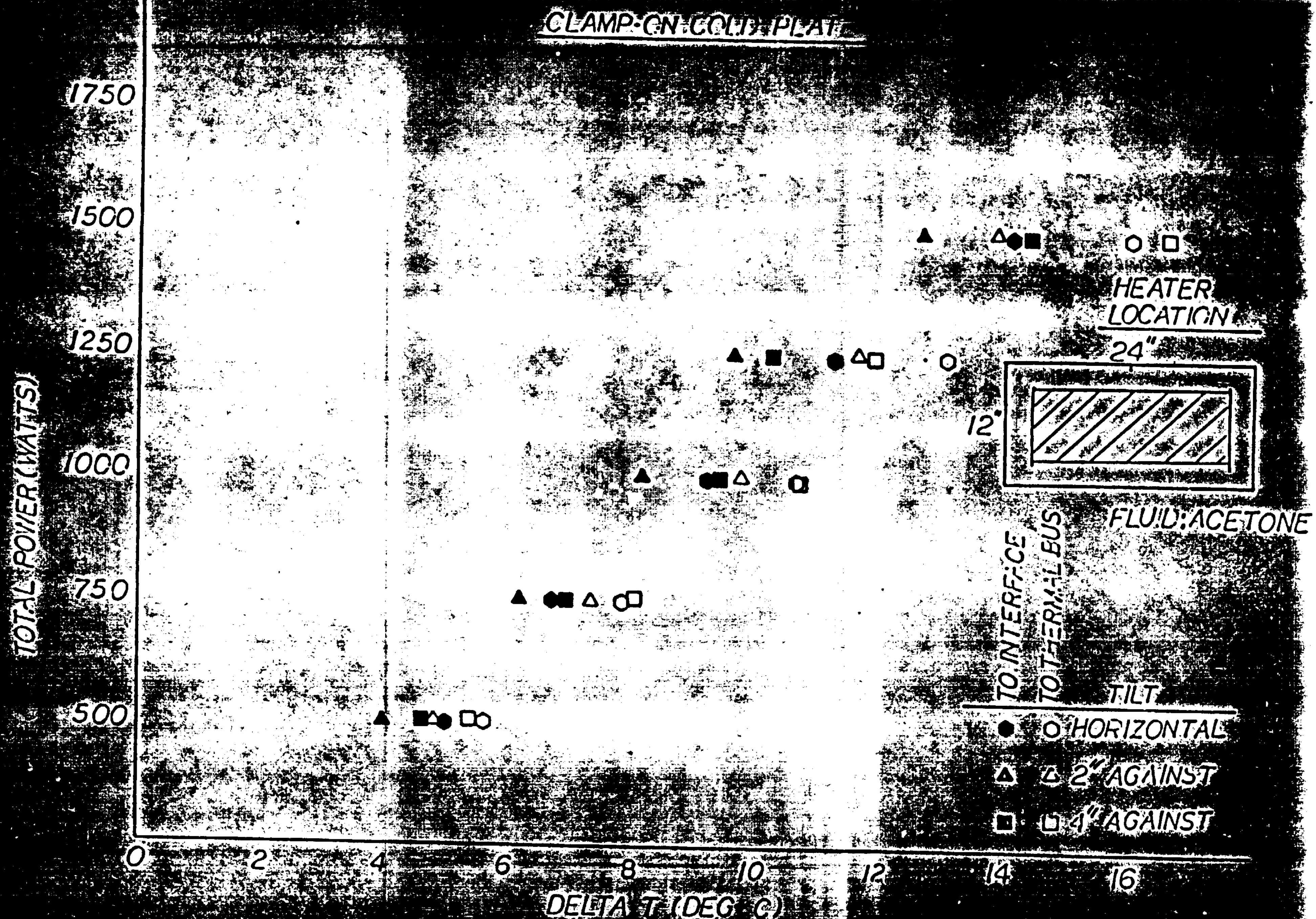

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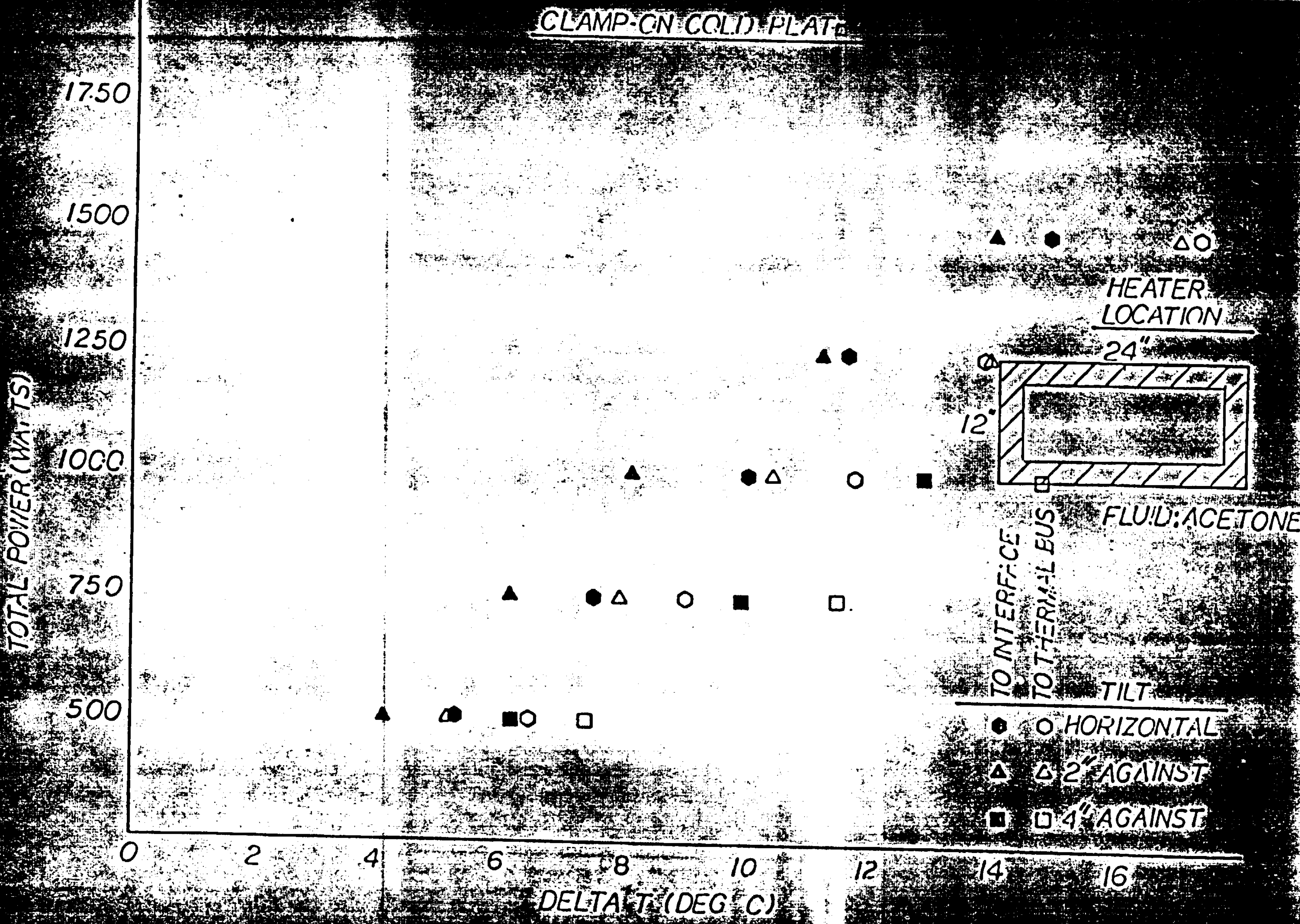

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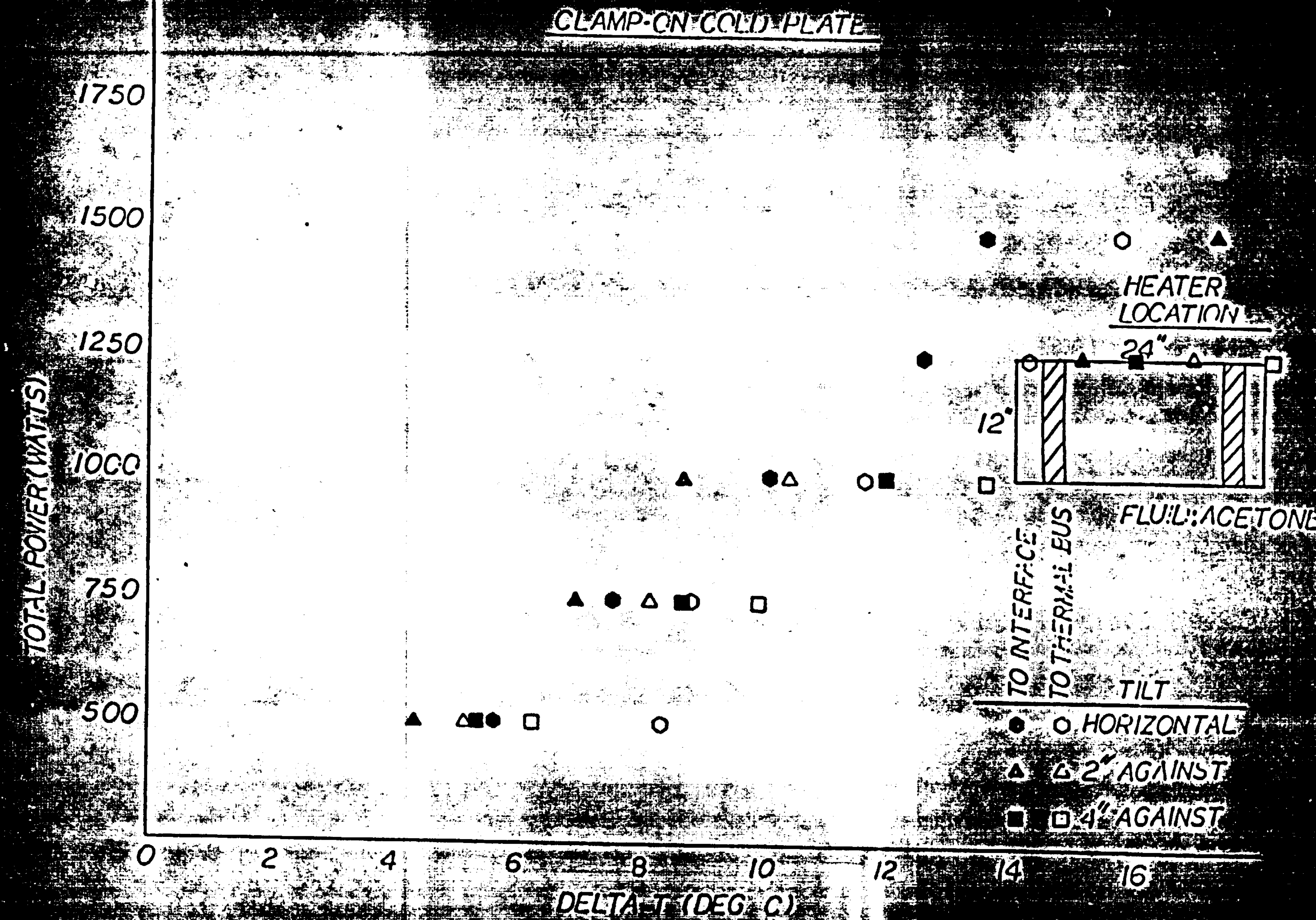

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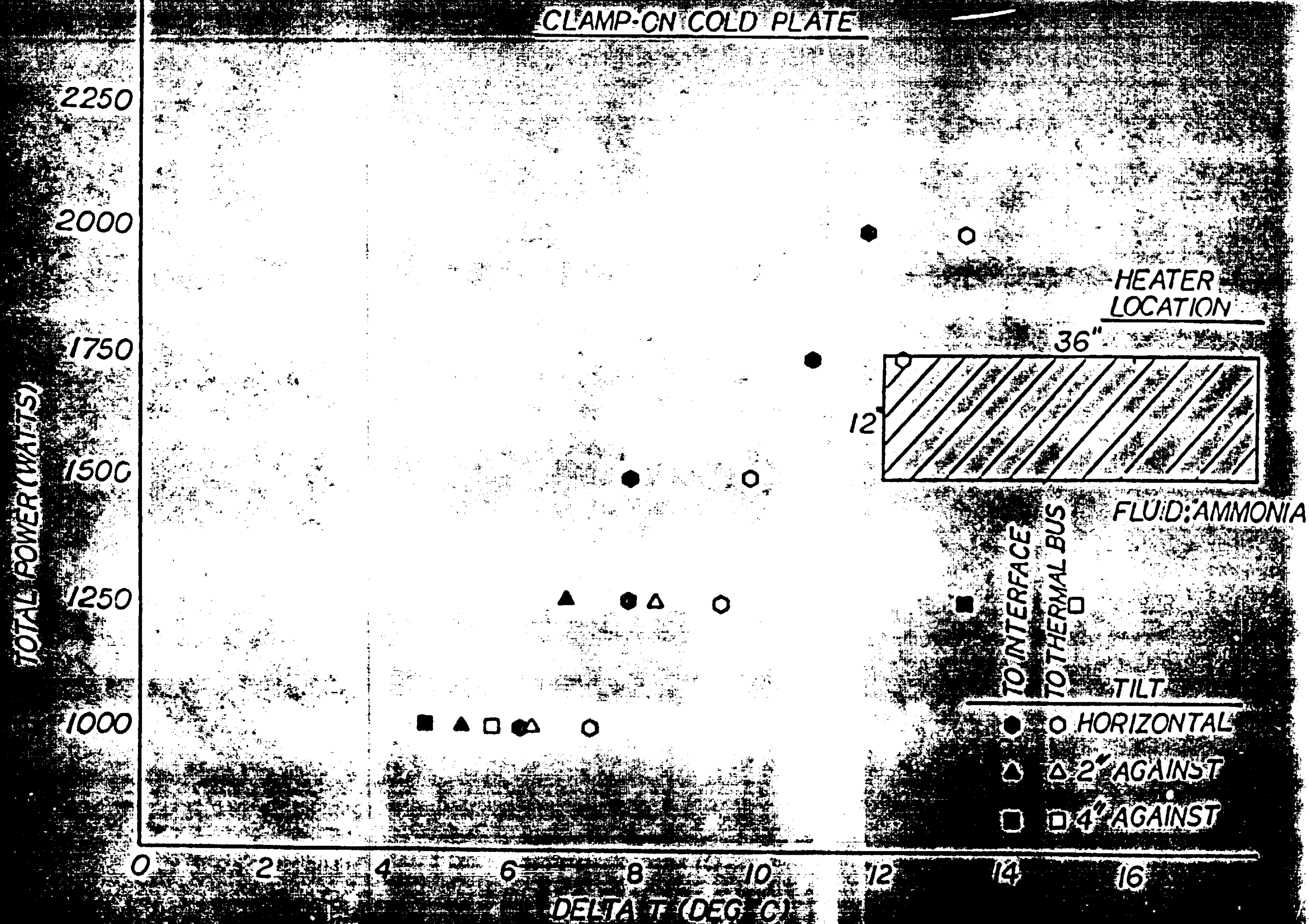

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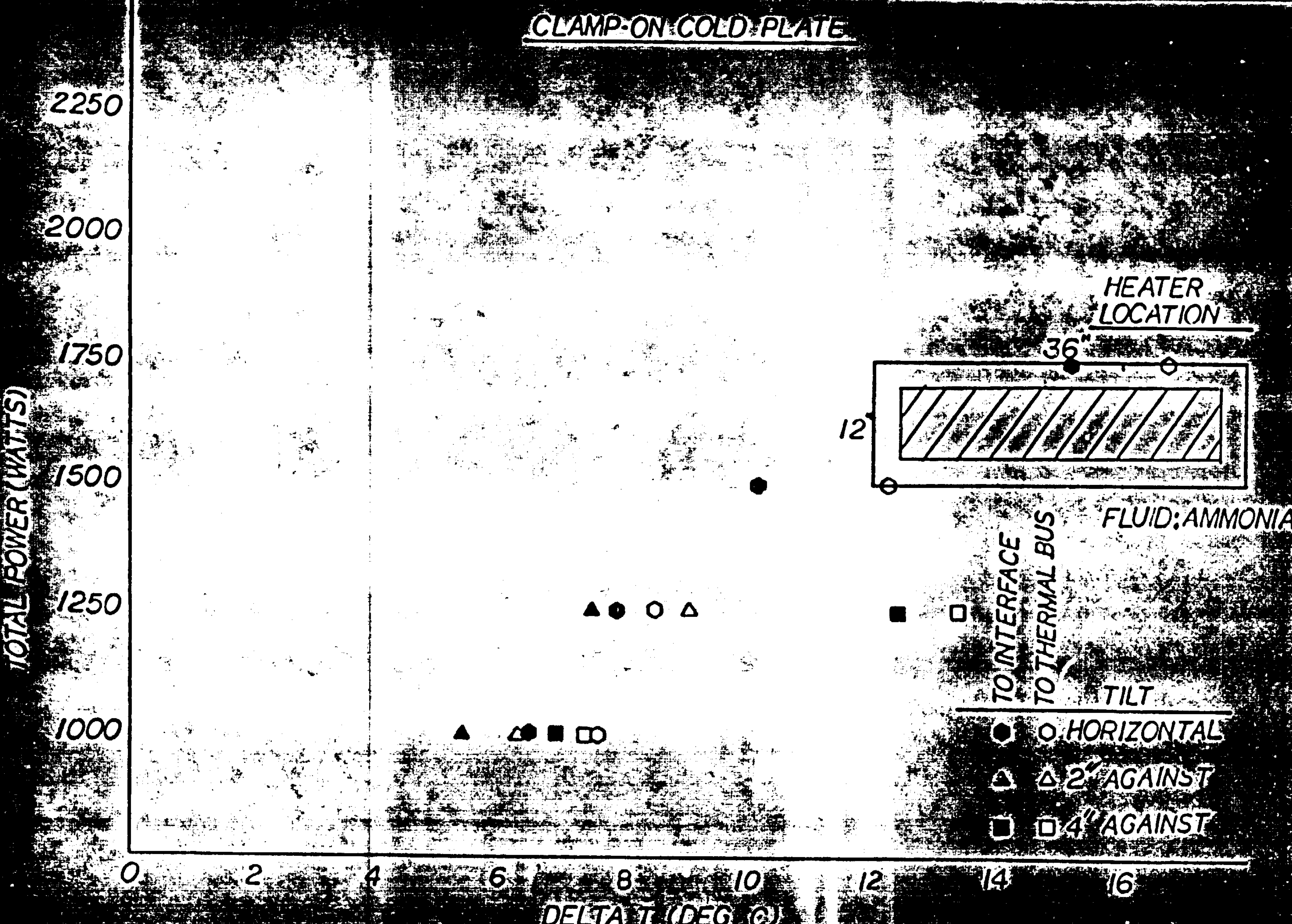

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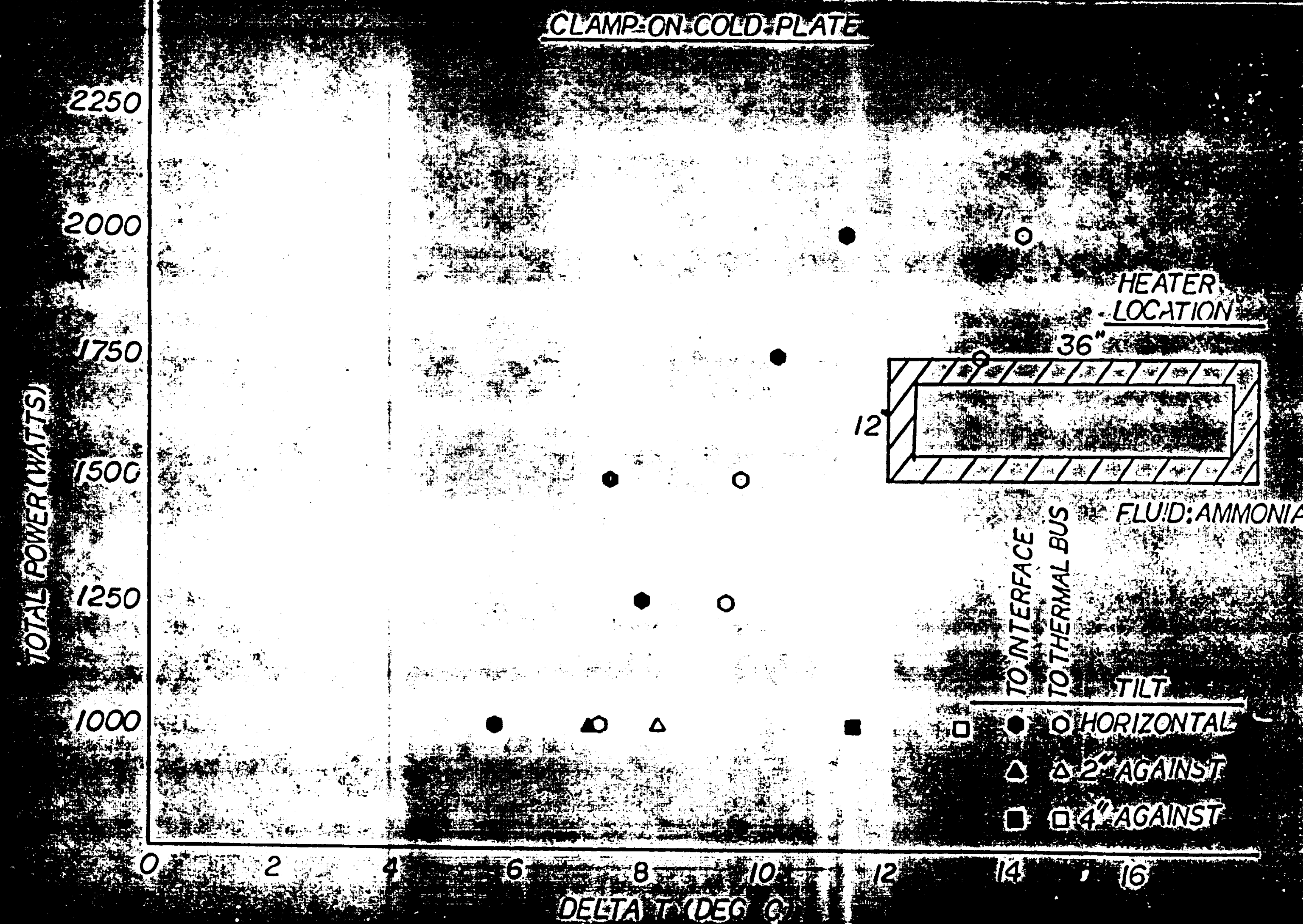

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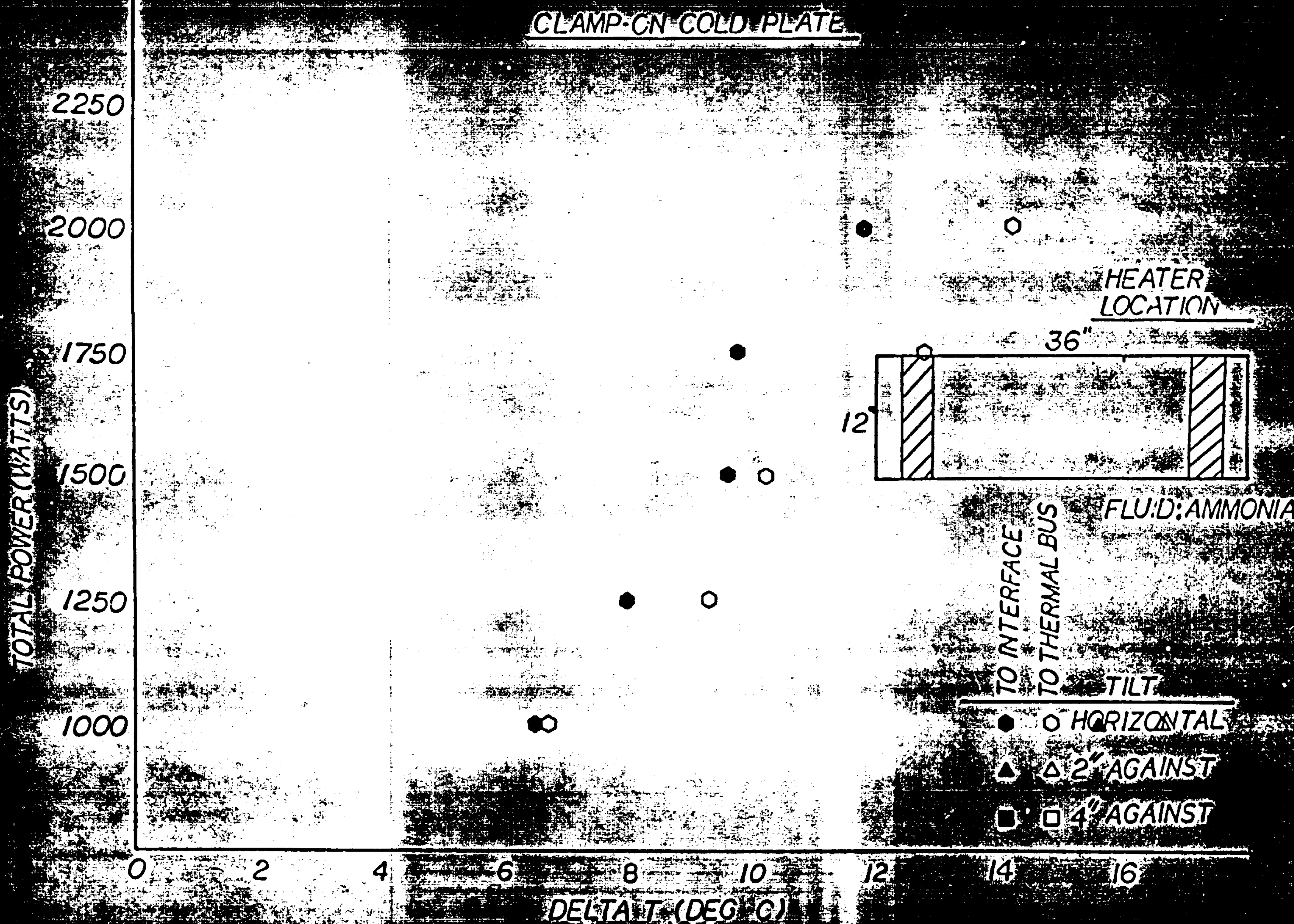

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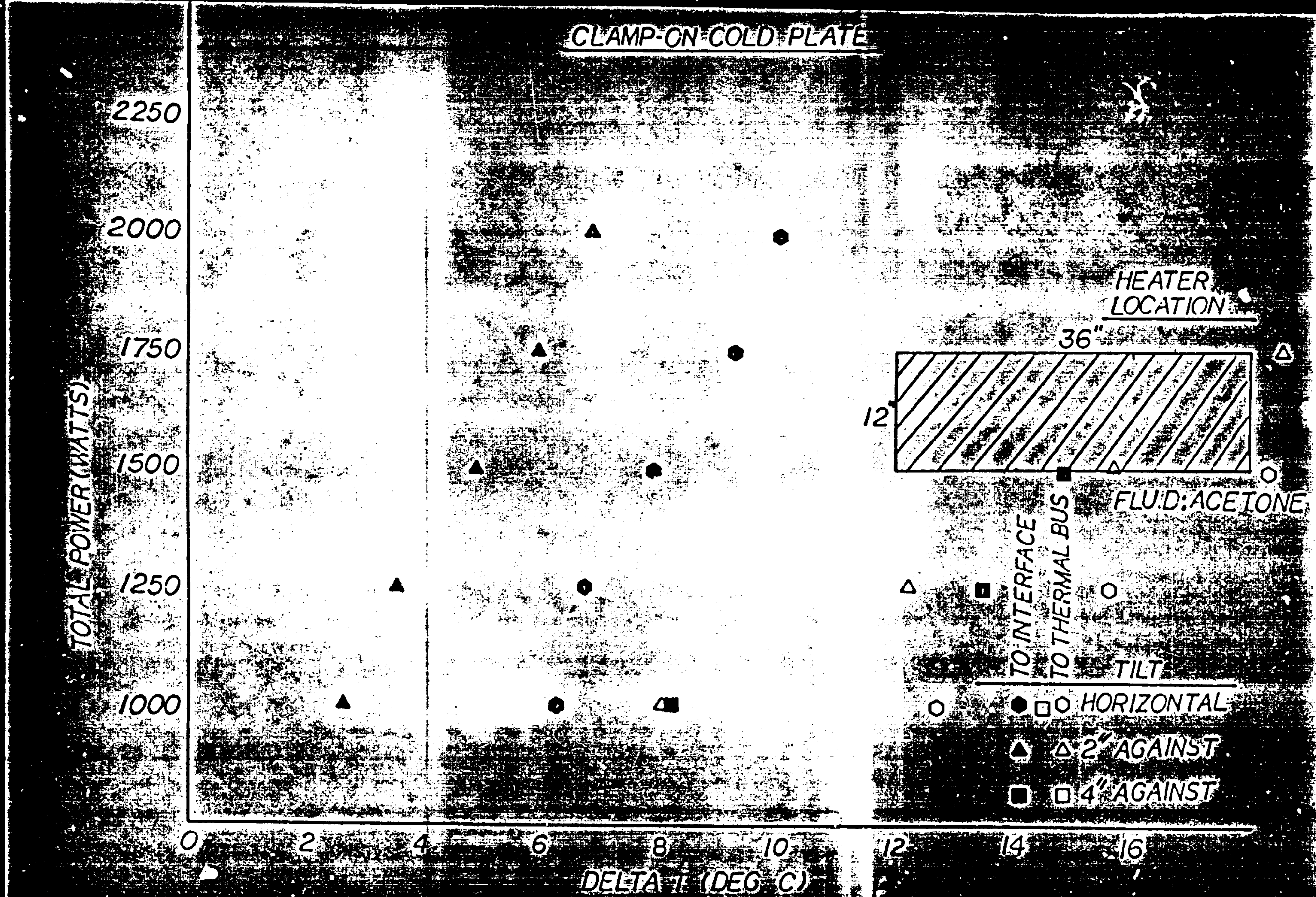

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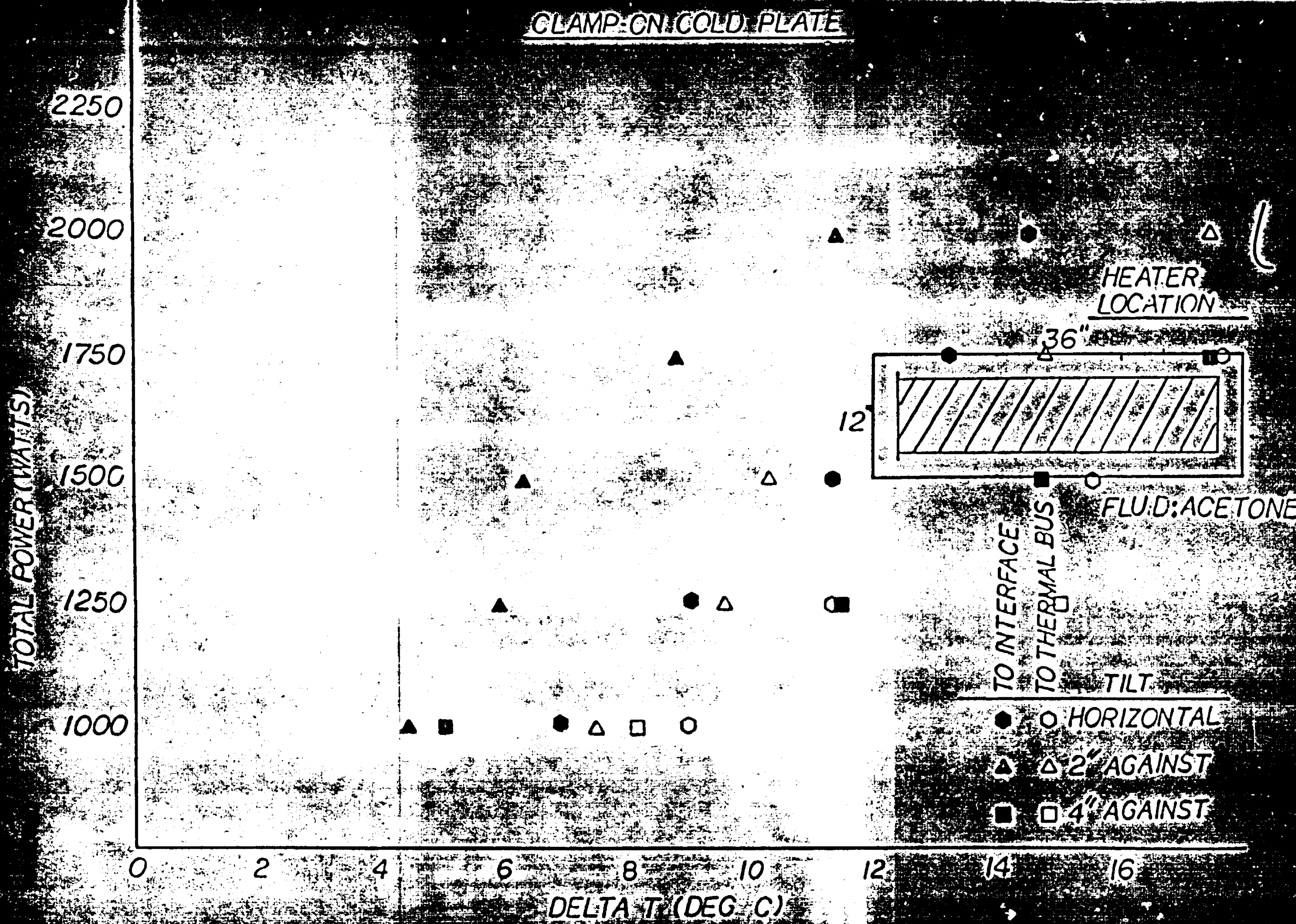

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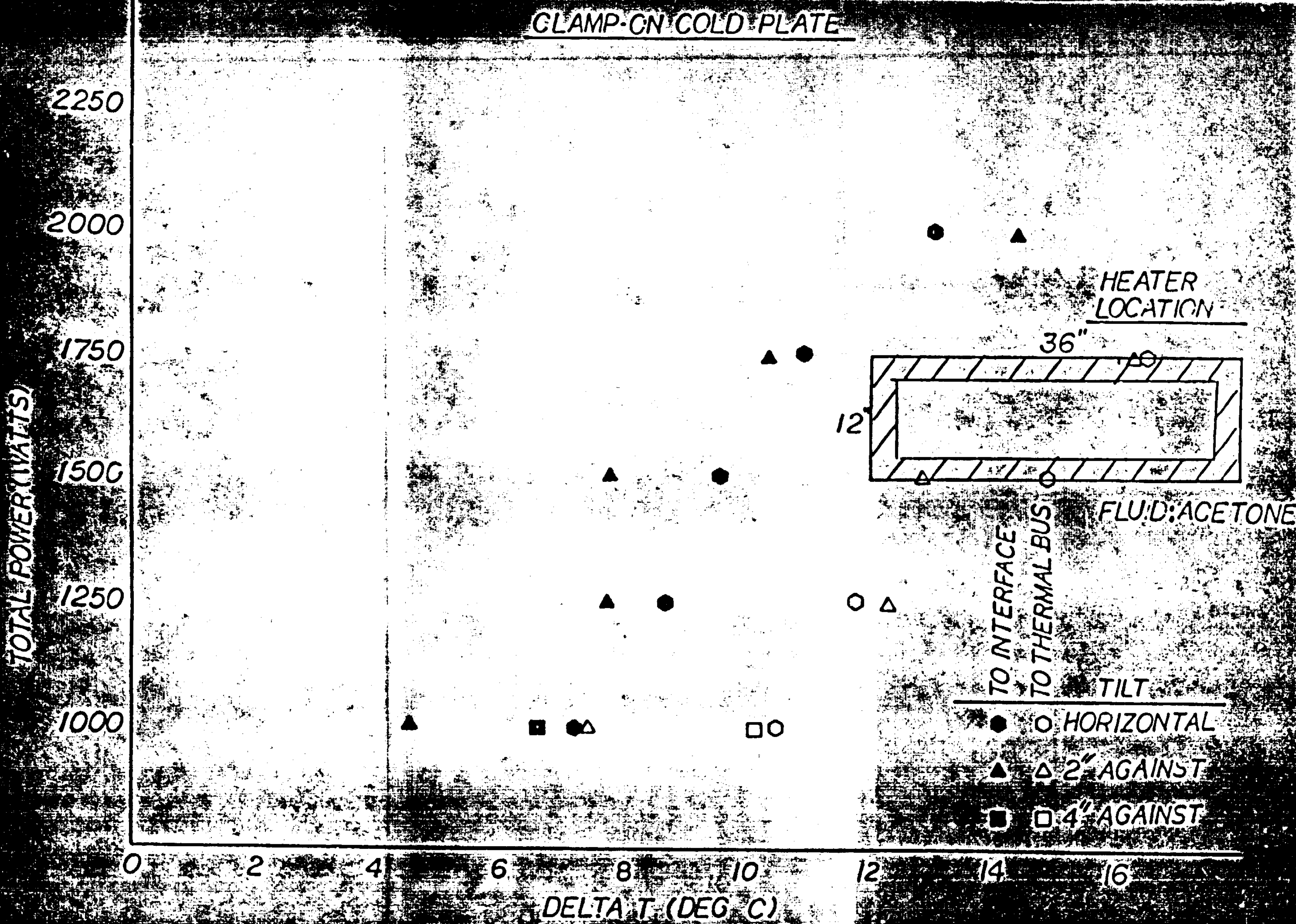

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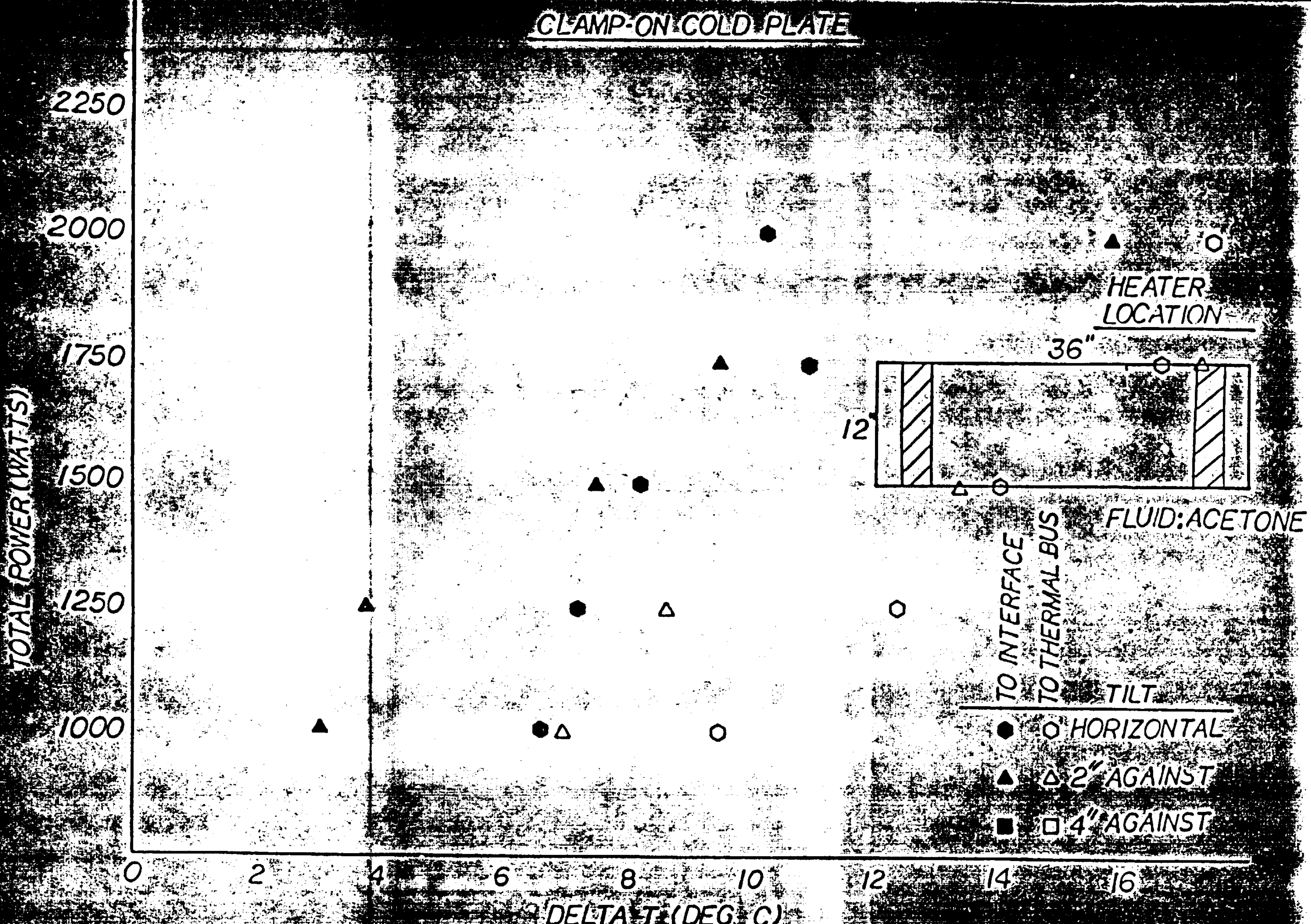

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